DESIGN OF MULTI-STAGE COMPRESSOR USING STREAMLINE CURVATURE METHOD

A Thesis

by

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ABSTRACT

Streamline curvature method is the most acceptable method used in compressor design. This method is superior to the simple radial equilibrium method because it accounts for losses and secondary flow in the computation.

A twenty-five stage subsonic axial compressor, with a total pressure ratio of 40.39, is designed using the streamline curvature method. This multi-stage compressor becomes part of an Ultra High Efficiency Gas Turbine (UHEGT). A number of design parameters are used to execute the mean line compressor modeling computation. The result of this computation becomes the input data in the streamline curvature method. The 3D blade profiler uses the streamline data (camberlines) to generate the actual blade profile. A solid modeling package is used to create 3D blades, a multi-stage compressor, and 3D final design of an Ultra High Efficiency Gas Turbine (UHEGT). The compressor performance map on designed rpm is generated. Finally, the blade pressure distribution is generated using CFD package software.

The streamline curvature method is successfully utilized for LP, IP, and HP compressor stage design. The computed total power needed for this multi-stage compressor is 91.2 MW. The inlet and outlet Mach numbers are 0.43 and 0.22, respectively. Based on the design parameters, the streamline curvature method gave lower flow deflection angles for LP compressor stages compared to IP and HP compressor stages.

DEDICATION

I dedicate this thesis to my lovely wife, Dr. Hj. Nur'aini, M. Ked (Ped), Sp. A. Thank you for all of your prayers, support, and encouragement. I would also like to dedicate this to my parents, sister, brother, and in laws.

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NOMENCLATURE

NACA	National Advisory Committee for Aeronautics
LP	Low pressure
IP	Intermediate pressure
HP	High pressure
l_m	Specific mechanical energy
H	Total enthalpy
h	Static enthalpy
P	Power
Po	Total pressure
P	Static pressure
V	Absolute velocity
W	Relative velocity
U	Rotational velocity
V_u	Circumferential velocity
V_m	Meridional velocity
$\dot{m}$	Mass flow rate
φ	Stage flow coefficient
λ	Stage load coefficient
r	Degree of reaction
μ	Meridional velocity ratio

ν	Circumferential ratio
α	Absolute flow angle
β	Relative flow angle
η	Isentropic efficiency
D_{eq}	Diffusion factor
D_m	Modified diffusion factor
r_c	Radius of curvature
T	Temperature
l	Computing station
s	Entropy
F_s	Body force
F_p	Pressure force
F_u	Circumferential force
F_n	Normal force
ρ	Density
ω	Angular velocity
BH	Blade height
A	Area
D_m	Mean diameter
t	Thickness
c	Camberline length
π	Pressure ratio

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1. INTRODUCTION

Streamline curvature method is widely used in turbomachinery industry for axial compressor design. This method is more realistic than simple radial equilibrium method. Assuming an axisymmetric flow, a simple radial equilibrium condition has constant meridional velocity and total pressure distributions with $rv_u = const$. This is so-called free vortex flow condition [1]. In practice, this condition is undesirable since secondary flow vortices predominate the flow field close to the blade hub and tip. On the other hand, streamline curvature method will consider this secondary flow by having the different value of meridional velocity and total pressure distribution from blade hub to tip. The streamline curvature method can be used in design or analysis by solving the radial equilibrium equation.

The computational streamline curvature program is used in this research. The air flow through compressor is assumed axisymmetric and inviscid. The fluid properties is also assumed ideal gas. The computational may include “through-blade” design approach, which calculate the distributed parameters inside a blade row. The preliminary design of compressor should be conducted beforehand to the level “across blade” design approach, which is the calculations based on blade’s leading and trailing edges [2][3].

The research will simulate one example of axial compressor design. The example is the main project of this research which is 25 stages subsonic axial compressor with pressure ratio = 40.4. NACA-65 blade base profile will be used in this subsonic compressor design. The preliminary design program will be executed beforehand in order to obtain the basic data as input to streamline curvature program. The goal of this research is to design multi-stage axial compressor using streamline curvature method. The compressor analysis will be included to show the compressor performance map and pressure distribution of blade profile.

2. LITERATURE REVIEW

The literature review is important to investigate the previous research of the streamline curvature method in multi-stage compressor design. The investigation provides the information needed for this research and how the compressor can be accurately designed.

2.1 Basic Compressor Design

The Compressor have numerous industrial applications wherever a greater working pressure is needed, or a forced airflow of any kind. For maximum efficiency, axial compressors are usually used. In general, a compressor stage starts with rotor and followed by stator. Figure 2.1 shows an axial compressor stage that consists of two rotors and one stator rows [1]. It also shows the velocity diagram of the compressor stage.

This research employs the multi-stage compressor that was comprised of three portions: low pressure (LP), intermediate pressure (IP), and high pressure (HP). In gas turbine engine, the compressor is usually followed by combustion chamber and turbine stages that will not be discussed in this research.

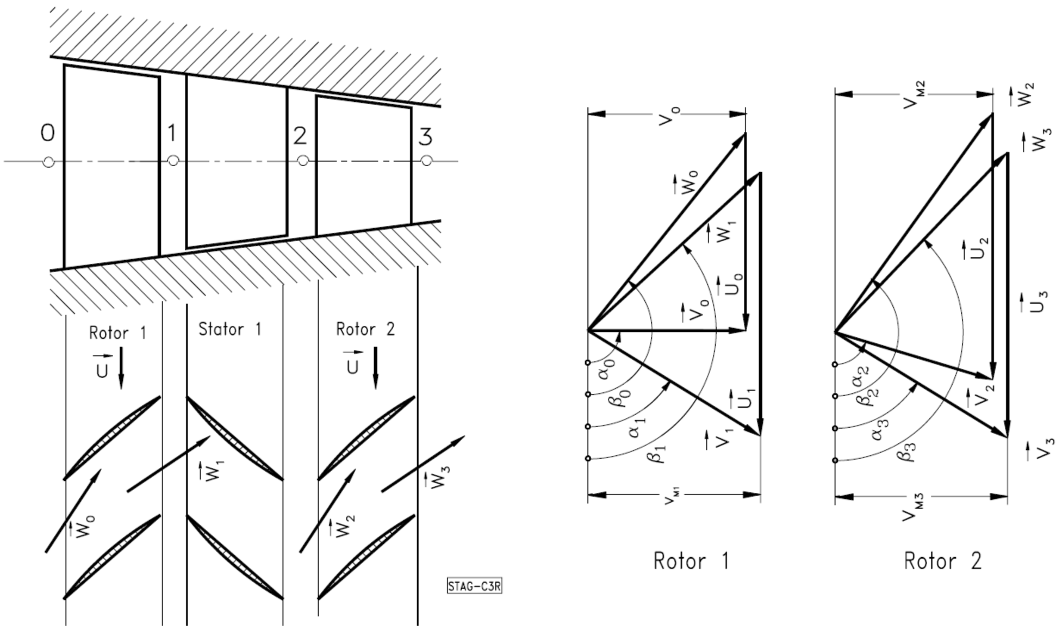

Figure 2.1. An axial compressor stage with rotor-stator-rotor configuration (left) and velocity diagrams for the first and second rotor (right) (from M. T. Schobeiri. "Turbomachinery Flow Physics and Dynamic Performance". 2nd edition [1]. with permission)

Air flows into the first rotor with absolute velocity in axial direction [1]. Since the rotating rotor working in relative frame of reference, the flow is deflected by the leading edge of rotor, so that the flow has relative velocity within relative system. At the trailing edge of rotor, the flow is again deflected, so that the relative velocity is decreased while the absolute velocity is increased. On the other hand, the process in the stator part is decreasing the absolute velocity. This compression process taking the mechanical energy input and converted into air potential energy in order to increase the air total pressure. This process causes the specific volume to decrease. In order to keep the same mass flow and axial velocity through the compressor, the cross section area needs to decrease in the compression flow direction.

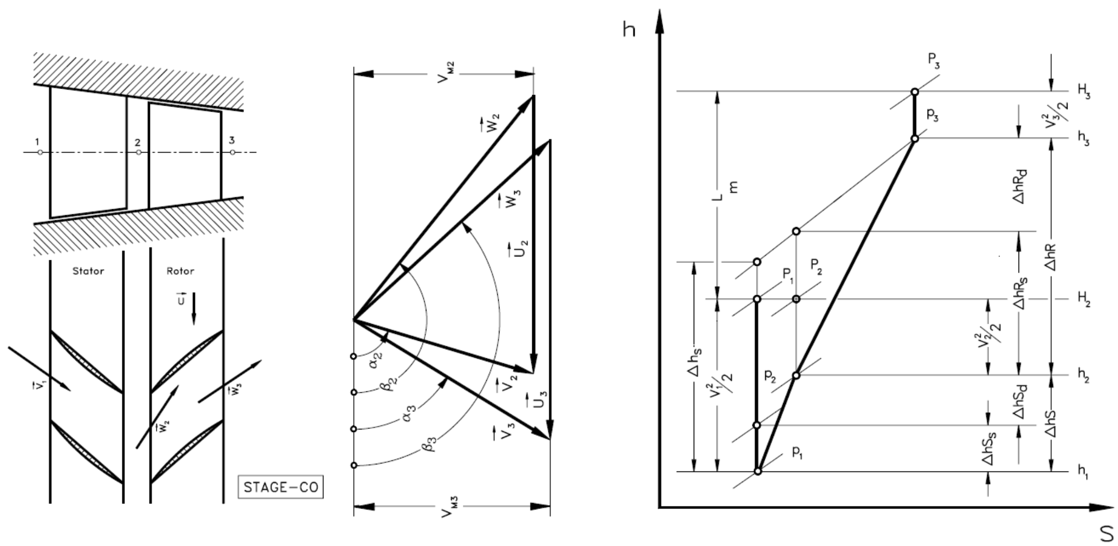

Figure 2.2. A compressor stage (left) with the velocity diagram (middle) and the compression h-s diagram (right) (from M. T. Schobeiri. "Turbomachinery Flow Physics and Dynamic Performance". 2nd edition [1]. with permission)

Figure 2.2 shows the velocity diagram and h-s diagram both for absolute system (stator) and relative system (rotor). In general, the compression process will cause the total pressure and total enthalpy to increase. This process consumes shaft power. We define the specific mechanical energy l_m which is equal to the difference between exit and inlet air total enthalpy. The value is positive since the energy is added to the compressor.

We define the specific mechanical energy l_m which is equal to the difference between exit and inlet air total enthalpy

$$l_m = H_1 - H_3 = \left(h_1 + \frac{1}{2}V_1^2\right) - \left(h_3 + \frac{1}{2}V_3^2\right) \qquad (2.1)$$

$$l_m = \frac{1}{2}[(V_2^2 - V_3^2) + (W_3^2 - W_2^2) + (U_2^2 - U_3^2)] \qquad (2.2)$$

Equation 2.2 is known as Euler turbine equation [4]. We also define power which is equal to mass flow rate $\dot{m}$ multiply by the specific mechanical energy l_m

$$P = \dot{m}U(V_{u1} - V_{u2}) \qquad (2.3)$$

The compression process also causes the specific volume to decrease, so based on continuity, we should decrease the cross section area.

We define stage flow coefficient as the ratio between meridional velocity component and circumferential velocity component

$$\varphi = \frac{V_{m3}}{U_3} \qquad (2.4)$$

and this parameter represents the mass flow behavior through the stage.

We define stage load coefficient as the ratio between the specific stage mechanical energy l_m and the exit circumferential kinetic energy U_3^2

$$\lambda = \frac{l_m}{U_3^2} \qquad (2.5)$$

This parameter represents the work capability of the stage. We also define stage degree of reaction r as the ratio between static enthalpy difference in rotor row and the static enthalpy difference in single stage.

$$r = \frac{\Delta h''}{\Delta h'' + \Delta h'} \tag{2.6}$$

this parameter represents the portion of energy transferred in the rotor blades row.

We define μ as meridional velocity ratio between the stator and rotor as follow

$$\mu = \frac{V_{m2}}{V_{m3}} \tag{2.7}$$

and we also define ν as circumferential ratio between the stator and rotor respectively

$$\nu = \frac{R_2}{R_3} = \frac{U_2}{U_3} \tag{2.8}$$

We can obtain the flow angle (as seen in velocity diagram from figure 2.3) based on the relation with dimensionless stage parameters as follow

$$\cot\alpha_2 - \cot\beta_2 = \frac{\nu}{\mu\phi} \tag{2.9}$$

$$\cot\alpha_3 - \cot\beta_3 = \frac{1}{\phi} \tag{2.10}$$

$$\lambda = \phi(\mu\nu\cot\alpha_2 - \cot\beta_3) - 1 \tag{2.11}$$

$$r = \frac{1}{2}\frac{\mu^2\phi^2 cot^2\alpha_2(\nu^2-1)-2\mu\nu\phi\lambda\cot\alpha_2+\lambda^2+2\lambda-\phi^2(\mu^2-1)}{\lambda} \tag{2.12}$$

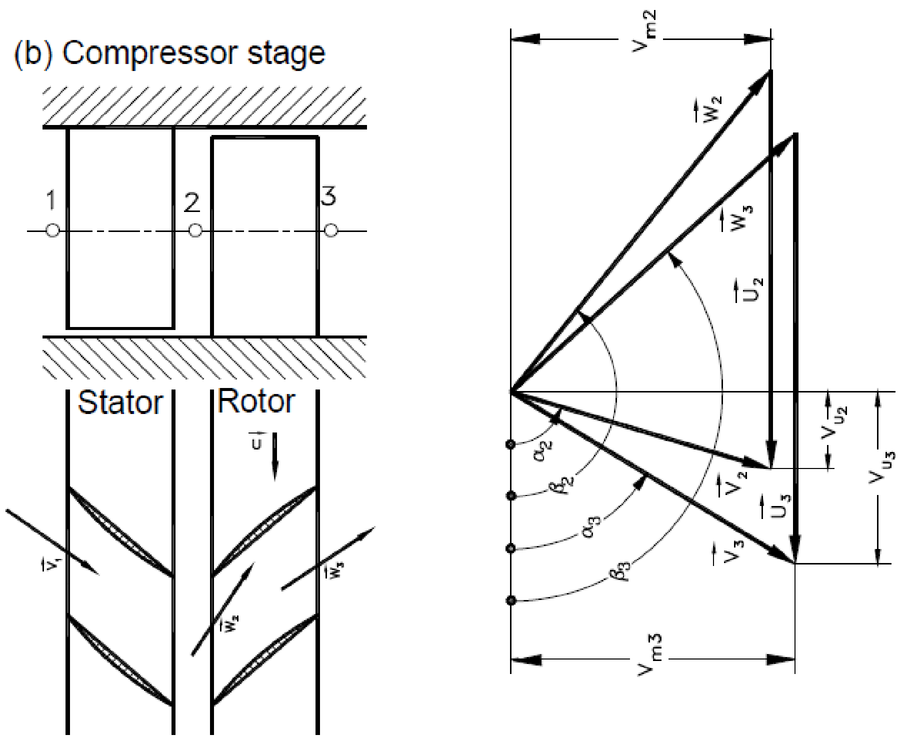

Figure 2.3. A compressor stage (left) with the velocity diagram (right) (from M. T. Schobeiri. "Turbomachinery Flow Physics and Dynamic Performance". 2nd edition [1]. with permission)

We define isentropic efficiency which is equal to the ratio between enthalpy difference in ideal compressor and the actual enthalpy change

$$\eta_{isen} = \frac{\Delta h_s}{\Delta h} \tag{2.13}$$

Ideal compressor has isentropic condition (adiabatic and reversible) which does not alter the entropy of flow gas. In actual condition, losses will generate an entropy rise. This will need higher enthalpy (or temperature) and work input to get the same pressure ratio as shown in figure 2.4.

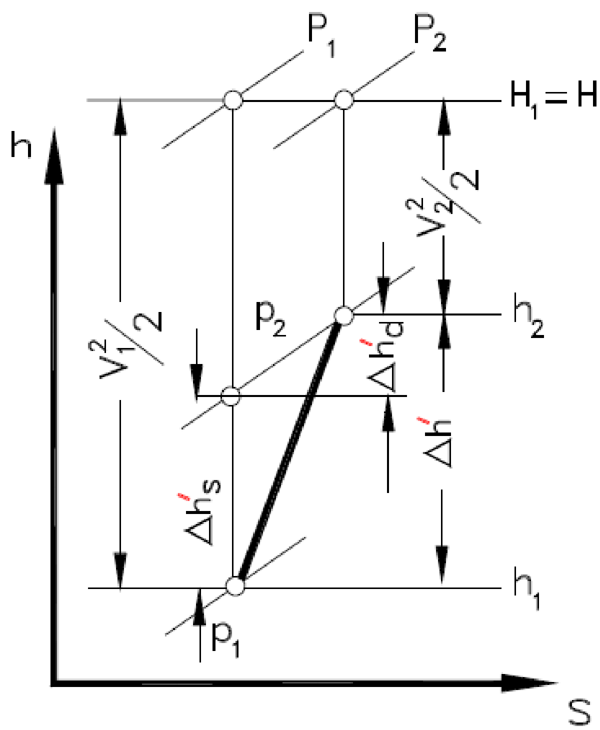

Figure 2.4. Compression process through a compressor stator row (from M. T. Schobeiri. "Turbomachinery Flow Physics and Dynamic Performance". 2nd edition [1]. with permission)

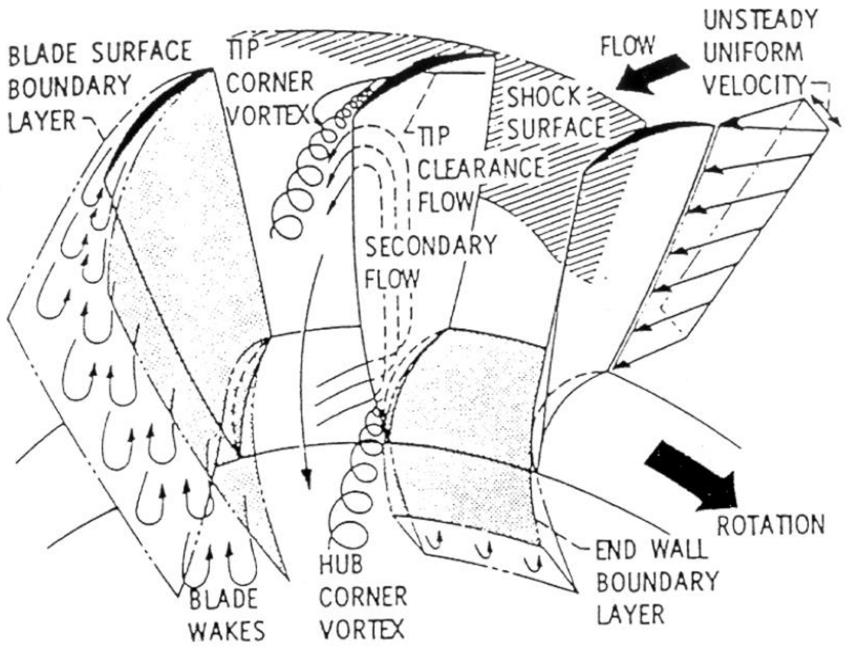

Figure 2.5. Losses generated in cascade blades

Figure 2.5 shows the major compressor losses are profile/primary losses, secondary flow (endwall hub and tip), and shock losses.

In the 1950s, Lieblein [5] developed correlation of diffusion ratio as loss parameter for subsonic compressor blades. Lieblien’s diffusion factor is defined as the ratio between maximum velocity and the outlet velocity of the blade.

$$D_{eq} = \frac{V_{max}}{V_2} = \frac{V_1}{V_2}\left(\frac{V_{max}}{V_1}\right) \qquad (2.14)$$

M. T. Schobeiri [1] developed correlation of total loss parameter as function of modified diffusion factor with immersion ratio as parameter as shown in figure 2.6.

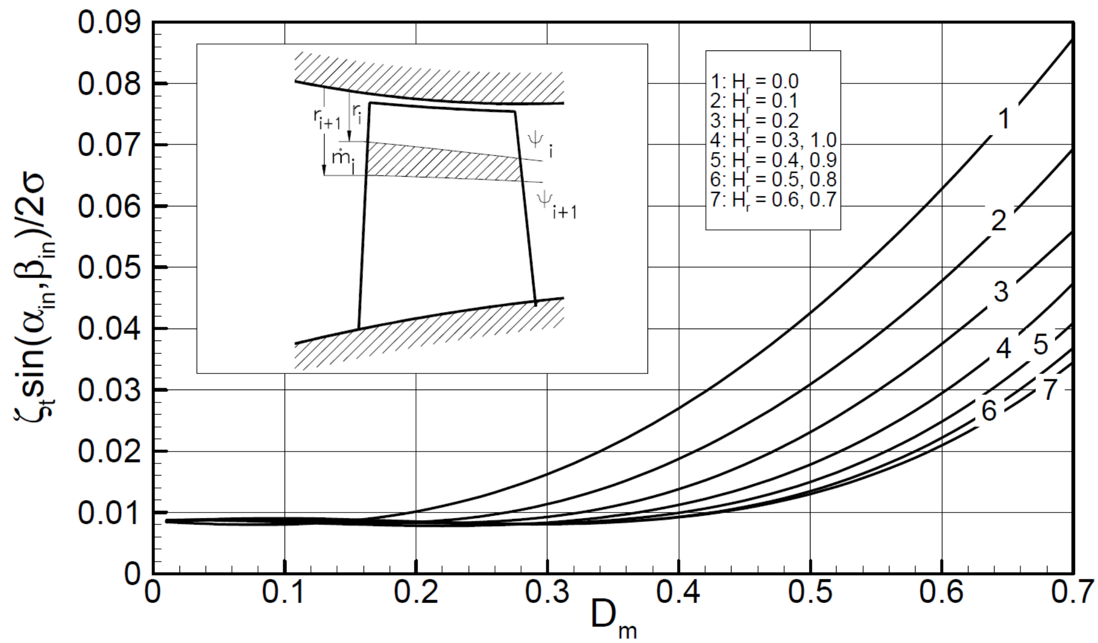

Figure 2.6. Total loss parameter with respect to modified diffusion factor with parameter of immersion ratio (from M. T. Schobeiri. “Turbomachinery Flow Physics and Dynamic Performance”. 2nd edition [1]. with permission)

2.2 Streamline Curvature Method

In compressor design, streamline curvature method is superior than simple radial equilibrium (free vortex flow) because it allow changing value of total pressure distribution and meridional velocity component from hub to tip of compressor blade. It consider the flow losses (primary losses, secondary losses, shock losses). It also calculates three-dimensional flow of stages, generating streamlines both in absolute and rotating frame of reference. Streamline curvature method can be used in design, off-design and analysis. In 1952, Wu [6] proposed a technique to solve three-dimensional flow field through a blade row by solving two-dimensional flow on the blade-to-blade surfaces and on the hub to tip surface. Vavra [7] explained the theoretical equation that could be used to obtain the streamline curvature equation. Novak [8] presented procedures in streamline curvature computation for fluid-flow problems. Aungier [9] presented quasi three-dimensional flow analysis that was based on Wu's technique [6]. Wennerstrom [2][3] presented the streamline curvature method and application that will be used in this research. Cumpsty [10] presented the application of streamline curvature method and showed the difference of meridional velocity in streamline curvature method and simple radial equilibrium. Korakianitis [11] presented through-flow analysis that reduced computational effort from streamline curvature method by taking into account the axial slope of the streamlines. Boyer [12] modified the streamline curvature method to improve the performance prediction in transonic axial compressor. Templaxis [13] developed 2D streamline curvature code to predict the performance compressor. Zhu [14] presented the off-design prediction performance of axial compressor by using streamline curvature method. Pachidis [15] presented the change of compressor performance with different radial pressure profile and pressure distortion using streamline curvature method.

We assumed the flow through the compressor is inviscid and axisymmetric. In streamline curvature method, we use intrinsic coordinates fixed in space along streamlines as presented in figure 2.7.

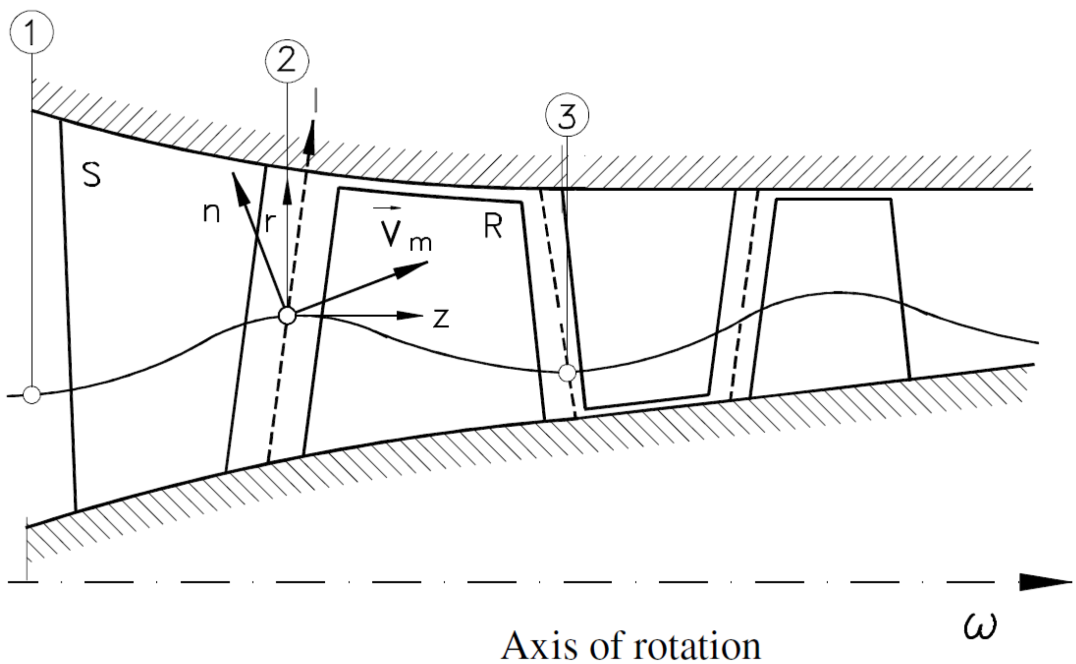

Figure 2.7. Flow through axial compressor, streamline, directions are: n = normal, m = meridional, r = radial, z = axial, and l = computing station (from M. T. Schobeiri. "Turbomachinery Flow Physics and Dynamic Performance". 2nd edition [1]. with permission)

In intrinsic coordinate system, we will pay attention for the flow along the streamline. Therefore, instead of using radial and axial coordinate system, we use meridional and normal direction, in addition to circumferential direction, to represent the flow component direction. We also need to define the basic geometry that is represented by computing station l as shown in figure 2.7. This computing station can be lies at leading or trailing edge of the blade, the space between stator and rotor, or at inlet or exit flow of the compressor. If the computing station is defined at the leading or trailing edge of the blade, it will be directly related with the basic shape of the rotor or stator blade.

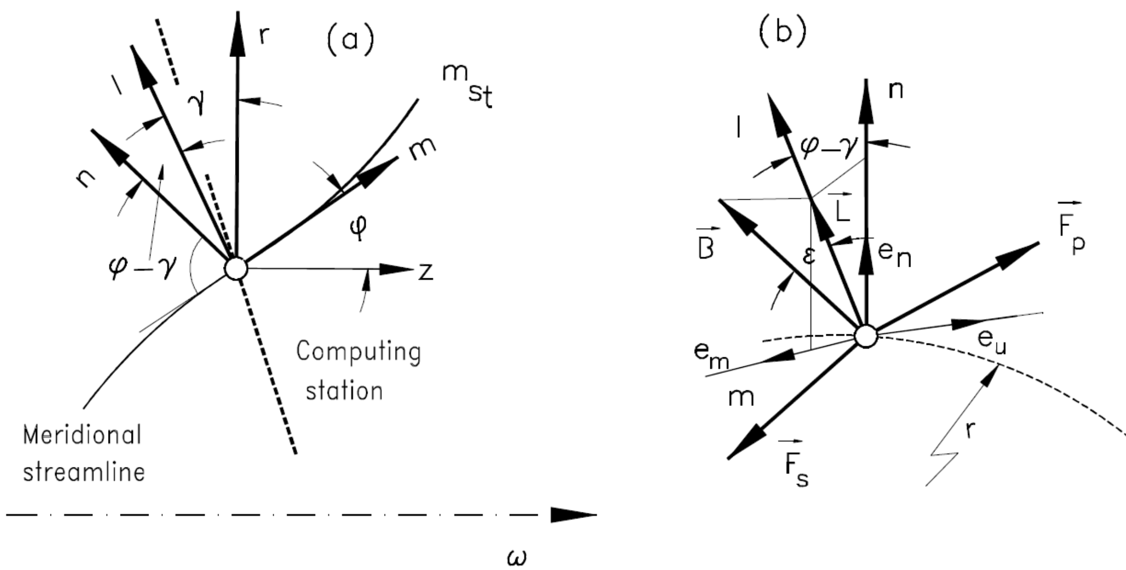

Figure 2.8. (a) Coordinate directions in the meridional plane, (b) orientation of vectors with respect to m, u, n orthogonal coordinate system (from M. T. Schobeiri. "Turbomachinery Flow Physics and Dynamic Performance". 2nd edition [1]. with permission)

Figure 2.8 illustrates the vectors involved in intrinsic coordinates system. Figure 2.8.a shows the coordinate direction in the meridional plane. We define stream surface slop angle φ as the angle difference between meridional and axial direction. The computing station lean angle γ is the angle difference between computing station l and radial direction. Figure 4.b describes the vectors of the streamsurface within the blade region. The variables m, u, and n are orthogonal axes of streamline coordinate system. F_s is the body force acting to the fluid and is represent the blade itself. The body force deflect the air flow at angle β (relative flow angle) in the m-u plane. The vector l represents the computing station and lies in the n-m plane. The vector B is tangent to the mean blade surface and is having blade lean angle ε from l direction and it lies in the l-u plane. F_p represent the force due to pressure across a blade and it is in the positive direction of rotation [3].

Momentum and energy equation are applied in the intrinsic coordinate system. Momentum equation in meridional (streamline) direction

$$V_m \frac{\partial V_m}{\partial m} - \frac{V_u^2}{r} \sin \varphi = -\frac{1}{\rho} \frac{\partial p}{\partial m} + F_m \tag{2.15}$$

and momentum equation in streamsurface normal direction

$$\frac{V_m^2}{r_c} - \frac{V_u^2}{r} \cos \varphi = -\frac{1}{\rho} \frac{\partial p}{\partial n} + F_n \tag{2.16}$$

and momentum equation in circumferential direction

$$\frac{V_m}{r} \frac{\partial (r V_u)}{\partial m} = F_u \tag{2.17}$$

We also use energy equation for steady adiabatic flow as follow

$$H = h + \frac{V^2}{2} \tag{2.18}$$

and the Clausius entropy relation

$$\frac{1}{\rho} dp = dh - Tds \tag{2.19}$$

while there is no velocity component in the n-direction, so we can decompose velocity to

$$V^2 = V_m^2 + V_u^2 \tag{2.20}$$

and we eliminate the derivatives with respect to the normal direction

$$\frac{\partial}{\partial n} = \sec(\varphi - \gamma) \frac{d}{dl} - \tan(\varphi - \gamma) \frac{\partial}{\partial m} \tag{2.21}$$

Those equation then are derived with respect to the computing station l-direction. The result is radial equilibrium equation in stationary absolute system as follow:

$$V_m \frac{dV_m}{dl} = \sin(\varphi - \gamma) V_m \frac{\partial V_m}{\partial m} + cos(\varphi - \gamma) \frac{V_m^2}{r_c} - \frac{V_u}{r} \frac{d(rV_u)}{dl} + \frac{dH}{dl} - T \frac{ds}{dl} - sin(\varphi - \gamma) F_m - cos(\varphi - \gamma) F_n \tag{2.22}$$

With streamline curvature method, meridional velocity V_m along each computing station (by l-direction) then is solved. RHS first term (fluid meridional acceleration) is

$$\sin(\varphi - \gamma)\, V_m \frac{\partial V_m}{\partial m} \tag{2.23}$$

and RHS second term (streamline curvature term):

$$cos(\varphi - \gamma)\frac{V_m^2}{r_c} \tag{2.24}$$

and RHS third term (gradient of angular momentum):

$$-\frac{V_u}{r}\frac{d(rV_u)}{dl} \tag{2.25}$$

and RHS forth term (enthalpy gradient):

$$\frac{dH}{dl} \tag{2.26}$$

and RHS fifth term (entropy gradient):

$$-T\frac{ds}{dl} \tag{2.27}$$

and RHS sixth term (meridional component of the blade force):

$$-sin(\varphi-\gamma)F_m \tag{2.28}$$

and also RHS seventh term (force normal to the streamsurface):

$$-cos(\varphi-\gamma)F_n \tag{2.29}$$

The absolute radial equilibrium can also be transformed into the rotating frame of reference by substitution of absolute circumferential velocity to relative circumferential velocity and absolute total enthalpy to relative total enthalpy. The resulting radial equilibrium in rotating frame of reference is as follow:

$$V_m \frac{dV_m}{dl} = \sin(\varphi-\gamma) V_m \frac{\partial V_m}{\partial m} + cos(\varphi-\gamma)\frac{V_m^2}{r_c} - \frac{W_u}{r}\frac{d(rW_u)}{dl} + \frac{dH_r}{dl} - T\frac{ds}{dl} - 2\omega W_u cos\gamma - sin(\varphi-\gamma)F_m - cos(\varphi-\gamma)F_n \tag{2.30}$$

Both of the above radial equilibrium equations are the basic of streamline curvature method. Those equation then can be modified for rotor or stator blade and blade-free space. In bladed region, the orthogonal component forces are composed of F_s (body force that oppose fluid motion) and F_p (force due to pressure difference across a blade) as follow

$$F_m = F_s \cos\beta - F_p \sin\beta \cos\varepsilon \tag{2.31}$$

$$F_u = F_s \sin\beta + F_p \cos\beta \cos\varepsilon \tag{2.32}$$

$$F_n = F_p[\cos\beta \sin\varepsilon \sec(\varphi-\gamma) + \sin\beta \cos\varepsilon \tan(\varphi-\gamma)] \tag{2.33}$$

where F_s, F_p, and F_u are obtained as follow

$$F_s = -cos\beta\, T \frac{\partial s}{\partial m} \tag{2.34}$$

$$F_p = \frac{F_u}{\cos\beta \cos\varepsilon} + \frac{\sin\beta}{\cos\varepsilon}\, T \frac{\partial s}{\partial m} \tag{2.35}$$

$$F_u = \frac{V_m}{r}\frac{\partial(rW_u)}{\partial m} + 2\omega V_m \sin\varphi \tag{2.36}$$

The resulting radial equilibrium after substitution in rotating frame of reference is as follow

$$V_m \frac{dV_m}{dl} = \cos^2\beta \left[(\sin(\varphi-\gamma) - \tan\varepsilon \tan\beta)V_m \frac{\partial V_m}{\partial m} + cos(\varphi-\gamma)\frac{V_m^2}{r_c}\right] - \cos^2\beta \left[V_m^2 \frac{\tan\beta}{r}\frac{d(r\tan\beta)}{dl} - 2\omega V_m(\tan\varepsilon \sin\varphi + \tan\beta \cos\gamma)\right] + \cos^2\beta \left[(\sin(\varphi - \gamma)\cos^2\beta - \tan\varepsilon \sin\beta\, cos\beta)T\frac{\partial s}{\partial m}\right] + \cos^2\beta \left[\frac{dH_r}{dl} - T\frac{ds}{dl} - V_m^2 \frac{\tan\varepsilon}{r}\frac{\partial(r\tan\beta)}{\partial m}\right] \tag{2.37}$$

In non-bladed region, the orthogonal component forces are composed of F_s (body force from entropy increase, e.g. swirl flow), while $F_p = 0$

$$F_m = -\cos^2\beta\, T \frac{\partial s}{\partial m} \tag{2.38}$$

$$F_u = -\sin\beta\, cos\beta\, T\frac{\partial s}{\partial m} \quad (2.39)$$

$$F_n = 0 \quad (2.40)$$

and the resulting radial equilibrium in non-bladed region is as follow

$$V_m\frac{dV_m}{dl} = \sin(\varphi-\gamma)\, V_m\frac{\partial V_m}{\partial m} + cos(\varphi-\gamma)\frac{V_m^2}{r_c} - \frac{W_u}{r}\frac{d(rW_u)}{dl} + \frac{dH_r}{dl} - T\frac{ds}{dl} - 2\omega W_u cos\gamma + sin(\varphi-\gamma)\cos^2\beta\, T\frac{\partial s}{\partial m} \quad (2.41)$$

The continuity equation is also used for each computing station

$$\dot{m} = \int_{r_h}^{r_t} V_m\rho\cos(\varphi-\gamma)2\pi r\, dl \quad (2.42)$$

with total enthalpy difference is related with angular momentum change

$$\Delta H = H_3 - H_2 = \omega(r_3V_{u3} - r_2V_{u2}) \quad (2.43)$$

3. OBJECTIVES

One of the main research purpose is designing subsonic multi-stage compressor with given basic parameters. Multi-stage compressor consists of 6 stages LP compressor, 9 stages IP compressor, and 10 stages HP compressor.

The research objectives that would be retracted from this research:

- To generate preliminary data for streamline curvature program
- To obtain streamline profile and blade profile for multi-stage subsonic compressor case.
- To generate the 3D solid modeling of multi-stage subsonic compressor.
- To obtain multi-stage compressor performance map and blades pressure distribution.

4. DESIGN PROCEDURE

This section provides all the information about the procedure in multi-stage compressor design using streamline curvature method.

- A number of parameters must be provided before designing a new compressor.
- The design parameters that could be specified are : mass flow, inlet pressure, pressure ratio, inlet temperature, rotational speed, stage degree of reaction, isentropic efficiency, stage flow coefficient, stage load coefficient, number of stages, solidity, stage and section spacing, hub diameter (if constant hub diameter) or mean diameter (if constant mean diameter compressor).
- The preliminary program that is based on mean line stream analysis will be executed and calculates stages pressure, enthalpy, density, temperature, entropy of thermodynamic properties of each stages.
- The preliminary program will iterate and calculate the mean diameter and blade height of each stages and relates with continuity equation

$$BH = \frac{A}{\pi\, D_m} = \frac{\dot{m}}{\rho V_m \pi\, D_m} \tag{4.1}$$

- The meanline absolute and relative velocity, mach number and angle distribution then will be calculated from preliminary program.

- Hub radius, tip radius, chord length, axial chord length, number of blades, stage spacing gap, and length of compressor will also be calculated from preliminary program.
- The preliminary program also calculates stage specific work and power of compressor stages.
- This research will use streamline curvature method in compressor design.
- In order to execute streamline curvature program, the user must supply the 2D blade and compressor stations geometry, and thermodynamic data of compressor stages.
- The stations geometry may be at leading / trailing edge of blade, inside of the blade, spacing between blades, or at inlet - outlet of compressor stages.
- The initial streamline pattern will be estimated by streamline curvature program by dividing the flow path into equal distance of each station.
- The slopes and curvature radius of streamline are calculated based on station by station data. Curvature radius are set to zero for the first and last stations.
- User specify the total pressure and total temperature at the first station.
- Initial meridional velocity will be estimated from inlet flow and first station data based on inviscid momentum equation.
- User specify total enthalpy and losses distribution from hub to tip for the stations of rotor blade.

- Exit flow angle and losses distribution data of stator blade discharge station should also be specified by user. The streamline curvature program then will calculate isentropic efficiency based on the losses data.
- The meridional velocity distribution of bladed region station will be computed based on rotor / stator data specified by the user and momentum equation.
- The achieved mass flow rate and flow gradient are calculated across all streamlines. The sign of flow gradient (positive or negative) indicate subsonic or supersonic condition. If user specify subsonic criterion, then the supersonic solution will not be valid.
- Mid-line meridional velocity is then re-estimated and iterated by

$$V_{m,new} = V_{m,old} + \frac{(\dot{m}_{specified} - \dot{m}_{achieved})}{\frac{d\dot{m}}{dV_m}} \qquad (4.2)$$

- These steps are repeated to the next station until the last computing stations.
- For blade-free space stations, the value of total enthalpy, entropy and angular momentum will be similar as the previous station.
- At each streamline, entropy is calculated from pressure loss or efficiency data. The density distribution will be iteratively calculated from the static pressure and entropy.

- For each streamline, the static pressure and entropy give the static enthalpy. The total and static enthalpies give the total velocity. The total velocity and circumferential velocity component will give the axial velocity component.
- The two criteria of convergence in streamline curvature program are: (1) The stream function for each streamline should be constant, (2) The meridional velocities at each point should not change.
- Relaxation factor is applied when relocating the streamlines.
- The final streamlines are generated by the program.
- The streamlines curvature program will generate numbers of 3D streamlines which reflect the hub-to-tip camberlines distribution of rotor and stator blades.
- Blade profiler program is then executed to generate blade profle from these camberlines (streamlines).
- For subsonic compressor, the blade profiler program employs NACA-65 thickness base profile.
- Blade profiler program calculate the length of 3D blade camberline and compute the thickness distribution of the actual profile as follow

$$\frac{t}{c} = \left(\frac{t}{c}\right)_{ref} \frac{\left(\frac{t}{c}\right)_{max}}{\left(\frac{t}{c}\right)_{\max ref}} \tag{4.3}$$

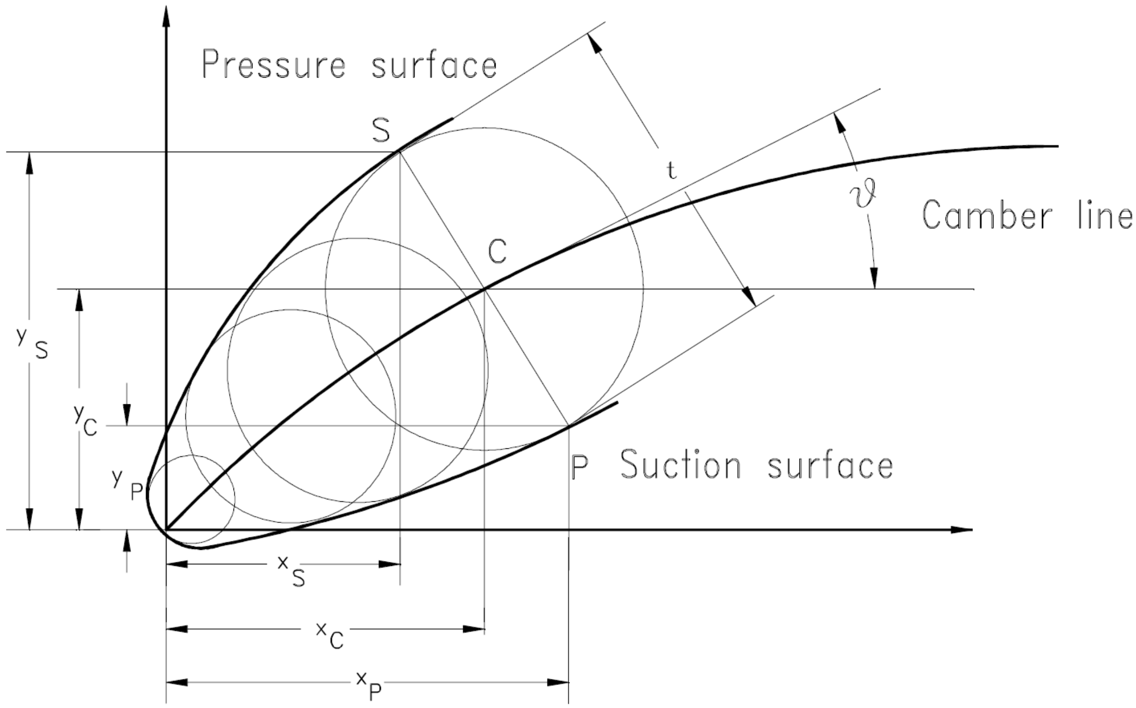

Figure 4.1. The superimposition of base profile on camberline (from M. T. Schobeiri. "Turbomachinery Flow Physics and Dynamic Performance". 2nd edition [1]. with permission)

- The superposition is executed by blade profiler program to the camberline on which the thickness is superimposed.
- The thickness superposition for suction side are

$$x = x_c - \left(\frac{t}{2}\right)\sin\vartheta \tag{4.4}$$

$$y = y_c + \left(\frac{t}{2}\right)\cos\vartheta \tag{4.5}$$

- The thickness superposition for pressure side are

$$x = x_c + \left(\frac{t}{2}\right) \sin\vartheta \qquad (4.6)$$

$$y = y_c - \left(\frac{t}{2}\right) \cos\vartheta \qquad (4.7)$$

- 3D CAD (Solidworks) is used to generate full rotor / stator blades based on hub-to-tip actual blade profile data.
- Solidworks is also used to create full 3D-compressor based from geometry data that is achieved from preliminary program and streamline curvature program. The multistage compressor then be coupled with combustor and turbine to create Ultra High Efficiency Gas Turbine (UHEGT).
- CFD packaged (Ansys Fluent) is used to generate the blade pressure distribution based on input flow data and blade profile.
- Design compressor performance map can be obtained by changing the inlet mass flow at designed rpm.

5. RESULTS AND DISCUSSION

5.1. Gas Turbine Performance and Compressor Basic Parameters

In this section, we will design multistage compressor that will be part of Ultra High Efficiency Gas Turbine. The design gas turbine will consists of 25 compressor stages, 3 combustion chamber, and 6 turbine stages. The h-s diagram of gas turbine cycle is shown in figure 5.1. The total compression pressure ratio generated by multi-stage compressor is 40.4. The thermal efficiency of gas turbine can reach 47%. This can be reached by employing three-stage of combustion in addition to high compression in multi-stage compressor.

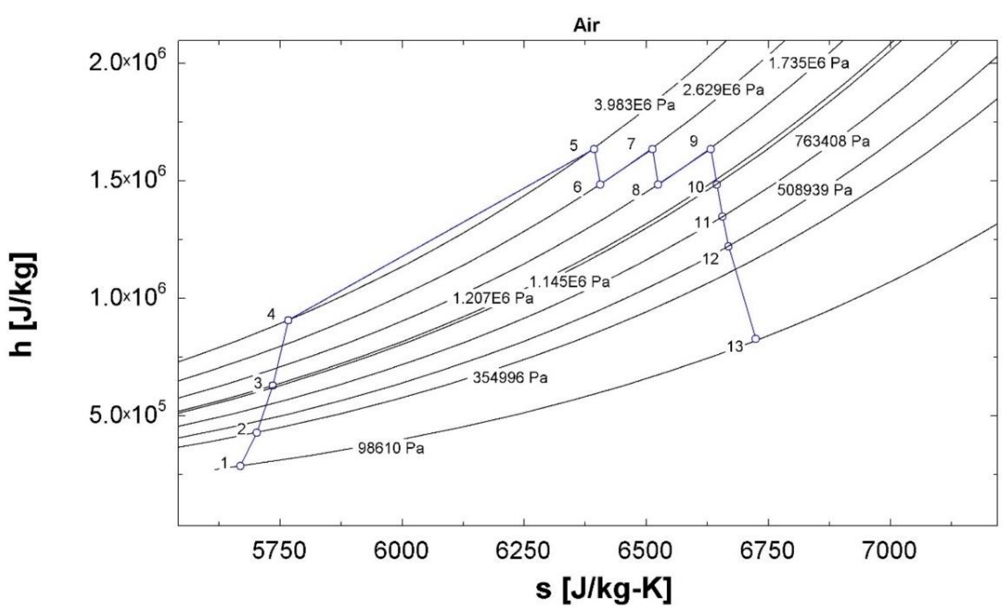

Figure 5.1. The enthalpy – entropy (h-s) diagram of Ultra High Efficiency Gas Turbine (UHEGT)

This research will focus on multi-stage compressor design for UHEGT with certain value of design parameters that shown in table 5.1.

Table 5.1. Design parameters of multi-stages compressor

Full Compressor	
Mass flow rate	150 kg/s
Inlet Pressure	98.61 kPa
Inlet Temperature	288.21 K
$\boldsymbol{\omega}$	5000 rpm
r	0.5
Stagger Angle	60 deg
Aspect Ratio	1/0.45
Chord/Spacing Ratio	1.3
Stage Spacing Ratio	0.15
Section Spacing Ratio	1.5
Low Pressure Compressor	
$\boldsymbol{D_{hub}}$	0.65 m
$\boldsymbol{\varphi}$	0.56
$\boldsymbol{\lambda}$	0.28
$\boldsymbol{\eta}$	92.49 %
$\boldsymbol{\pi}$	3.6
$\boldsymbol{n_{stg}}$	6
Intermediate Pressure Compressor	
$\boldsymbol{D_{hub}}$	0.58 m
$\boldsymbol{\varphi}$	0.58
$\boldsymbol{\lambda}$	0.28
$\boldsymbol{\eta}$	91.03 %
$\boldsymbol{\pi}$	3.4
$\boldsymbol{n_{stg}}$	9
High Pressure Compressor	
$\boldsymbol{D_{hub}}$	0.54 m
$\boldsymbol{\varphi}$	0.6
$\boldsymbol{\lambda}$	0.28
$\boldsymbol{\eta}$	90.83 %
$\boldsymbol{\pi}$	3,3
$\boldsymbol{n_{stg}}$	10

5.2. Results of Mean Line and Streamline Curvature Method

The preliminary program is then executed and compute thermodynamic properties of each stage. This calculation is based on mean line parameter of compressor stages. The streamline curvature program then executed and compared the results. Figure 5.2 shows the temperature and pressure distribution of multi-stage compressor that is generated by the preliminary program and streamline curvature program. The LP, IP, and HP stages have different slope of pressure distribution due to different pressure ratio between these compressor stages.

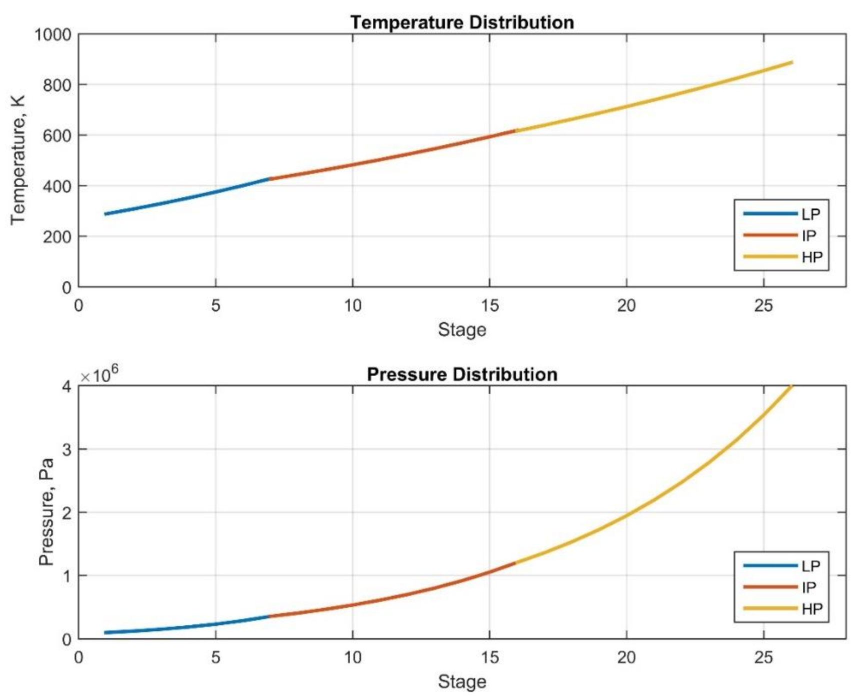

Figure 5.2. Temperature and pressure distribution of multi-stage compressor

The mean line preliminary program also give 2D basic geometry of the compressor stages. These geometry then are used as input data of streamline curvature program. Figure 5.3 shows the blade height and tip / hub radius distribution of multistage compressor that is generated by the preliminary program and used by streamline curvature program as the input stations data. These station may be a straight lines or have slopes that are confined by hub and tip radius.

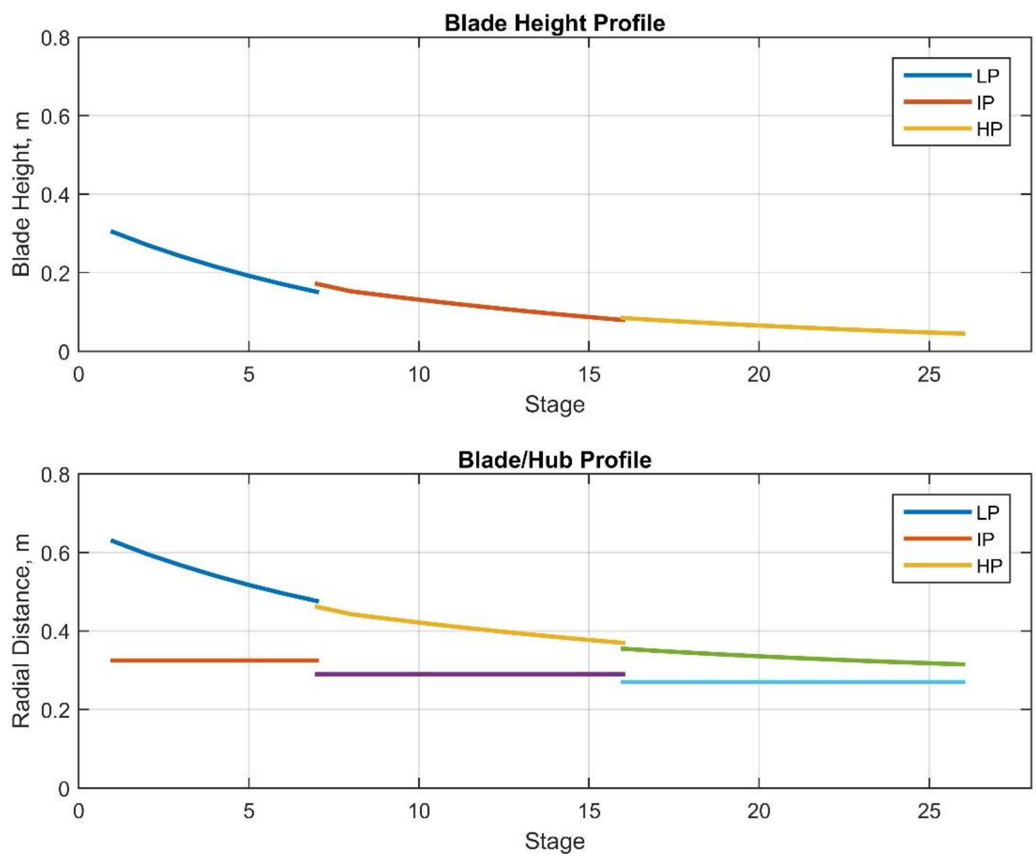

Figure 5.3. Blade height and blade/hub profile of multi-stage compressor

The preliminary program and streamline curvature program also can generate the enthalpy and Mach number distribution of multistage compressor. Figure 5.4 shows enthalpy and mach number distribution based on mean line diameter of compressor stages.

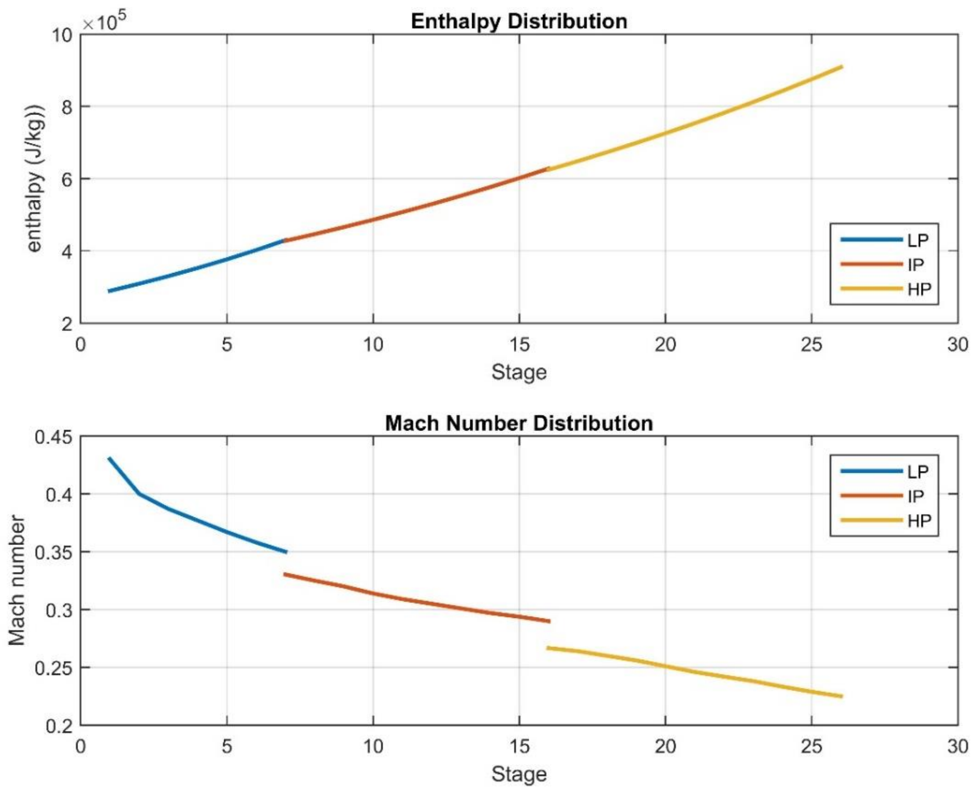

Figure 5.4. Enthalpy and mach number distribution of multi-stage compressor

Both the preliminary program and streamline curvature program can calculate the power needed for multi-stage compressor. The computed LP needed power = 20.74 MW.

The computed IP needed power = 29.25 MW. The computed HP needed power = 41.21 MW. So, the total needed power = 91.2 MW.

As input data for streamline curvature program, we specify 2D geometry /stations of LP, IP, and HP compressor based on the result geometry of preliminary program. Stations are placed at leading/trailing edge of blades, space between the blades, and at inlet and outlet of compressor stages. Stations for LP and IP stages are designed with slopes, in order to maintain the length of blade chord from hub to tip. Stations for HP are designed in straight radial direction, since there is no significant deviation of blade chord from hub to tip. We specify rotor enthalpy distribution for streamline curvature program based on the mean line enthalpy.

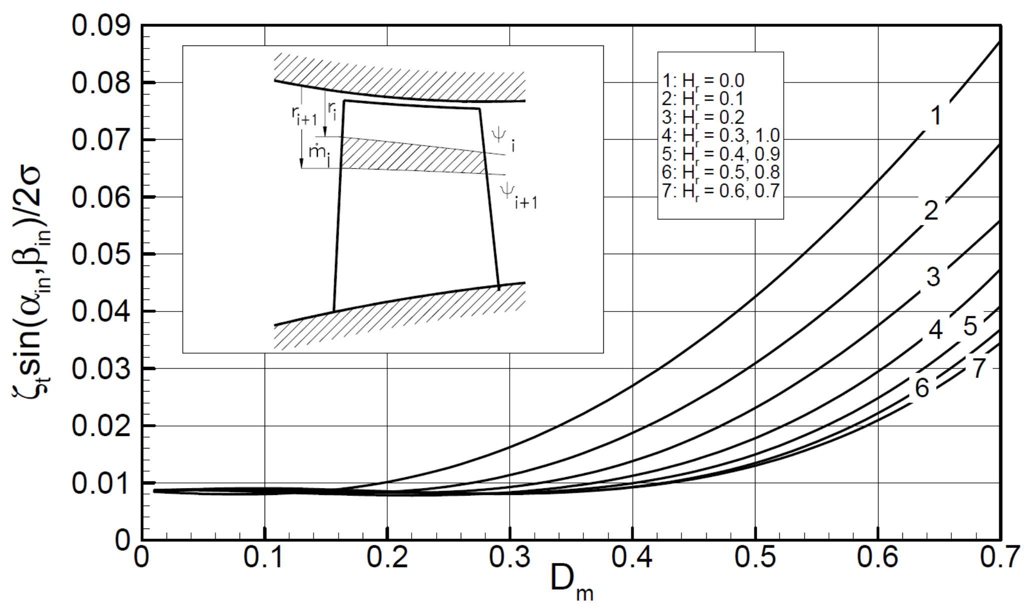

Figure 5.5. Total loss parameter with respect to modified diffusion factor with parameter of immersion ratio (from M. T. Schobeiri. "Turbomachinery Flow Physics and Dynamic Performance". 2nd edition [1]. with permission)

We also need to give losses as input data for streamline curvature program. Losses are specified in every station at inside or at edge of rotor or stator blades. These losses are obtained from the figure 5.5 by specifying Dm $_{LP}$ = 0.4, Dm $_{IP}$ = 0.5, and Dm $_{HP}$ = 0.6. The streamline curvature program then will calculate isentropic efficiency of multi-stage compressor based on these losses and will be compared with the isentropic efficiency that already specified on basic design parameter. Figure 5.6 shows the example of losses distribution for HP compressor stages.

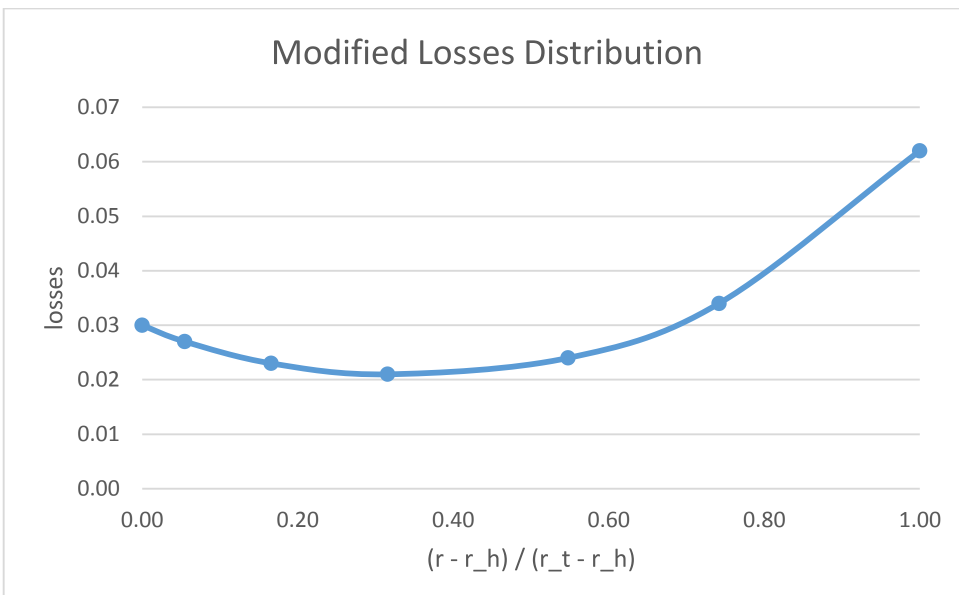

Figure 5.6. Modified losses distribution from hub to tip of HP blades

The streamline and geometry profile for LP, IP, and HP compressors will be generated by streamline curvature program as shown in figure 5.7 for LP stages, figure

5.8 for IP stages, and figure 5.9 for HP stages. These figures also show the stations at rotor and stator leading edge and trailing edge, and also stations at spaces between the blades. We can also observed that we only use slope station for LP and HP stages. The generated 3D streamlines are basically will be used as camberlines for the rotor and stator blades. The stator camberlines are generated using radial equilibrium formula in absolute frame of reference, while the rotor camberlines are generated in rotating frame of reference.

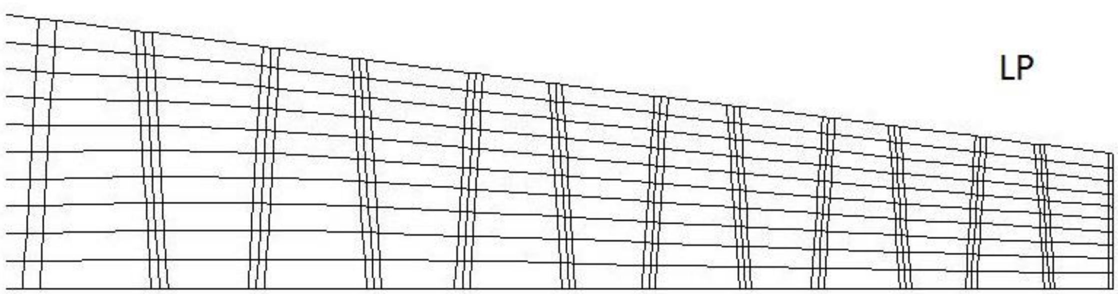

Figure 5.7. Streamline and geometry profile for LP compressor

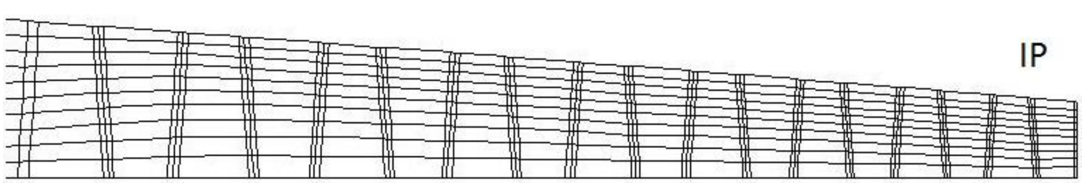

Figure 5.8. Streamline and geometry profile for IP compressor

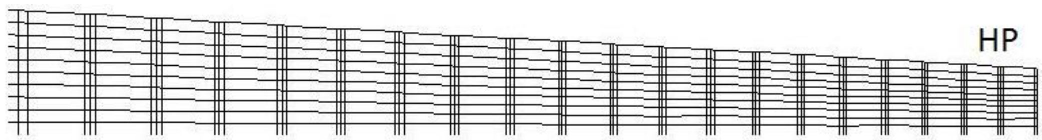

Figure 5.9. Streamline and geometry profile for HP compressor

5.3. Compressor 3D Modeling

We use the blade profiler program to generate the actual blade profile based on the rotor & stator camberlines that are generated from streamline curvature program. We use 11 blade profile from hub to tip that are based on 11 streamlines. Figure 5.10 shows the example of rotor blade profile from LP stage #1 in side view and top view. We can observed that in spite of the station profile has a slope, the chord length from hub to tip have relative negligible deviation.

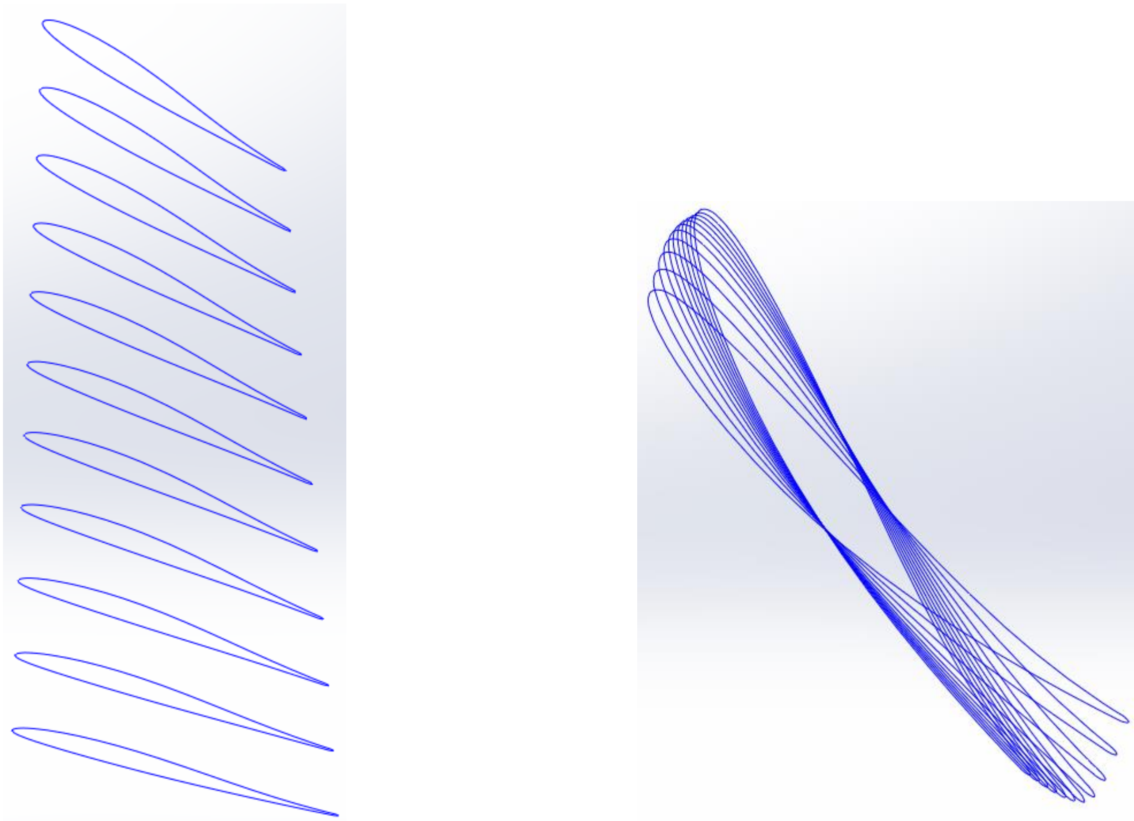

Figure 5.10. Side view (left) and top view (right) of actual blade profile for LP rotor stage #1 generated from blade profiler

Figure 5.11 shows the actual blade profile of stator blade for LP stage #1 in side view and top view. It is also generated based on 11 streamlines from streamline curvature program. We also can observed that the chord length from hub to tip are not much varied in spite of the stator station profile have slopes. The only observable thing in the stator blade profile is the variation of flow angle or the twisting.

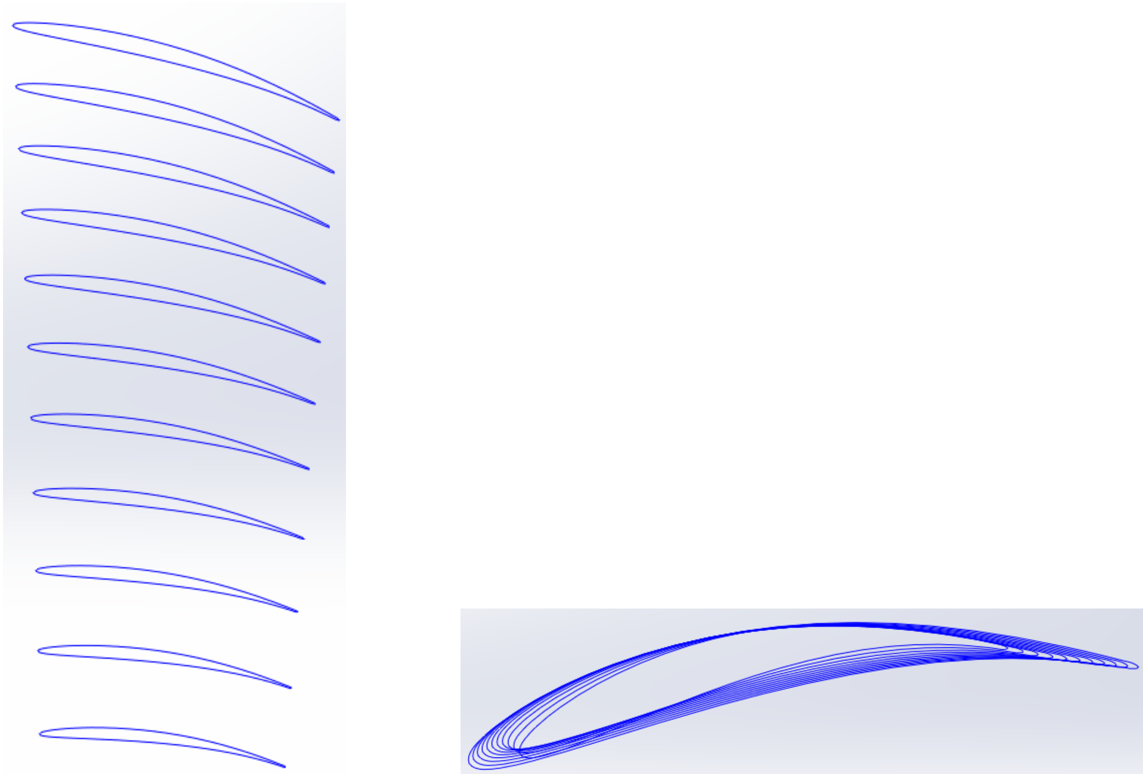

Figure 5.11. Side view (left) and top view (right) of actual blade profile for LP stator stage #1 generated from blade profiler

We use 3D CAD (Solidworks) to generate actual blade based on the rotor & stator blade profiles that are generated from blade profiler program. It is generated from 11 blade profiles from hub to tip for every rotor and stator blades. Figure 5.12 shows the examples of actual rotor and stator blade of LP stage #1. We can observed that the the rotor and stator blades have the twisting and the lean angle from hub to tip. The rotor and stator blades also have root or crown on purpose, so that the blades can be easily assembled into shaft or casing.

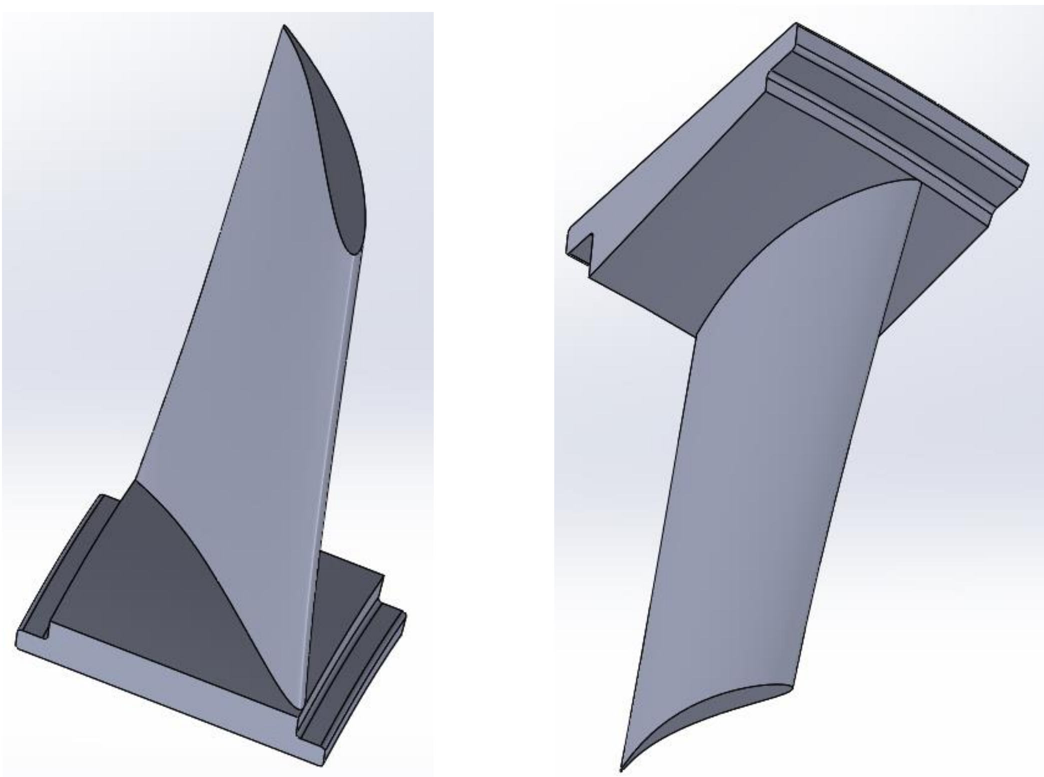

Figure 5.12. Rotor blade (left) and stator blade (right) of LP stage #1

We can generate the cascade midline blade profile using the blade profile #6. It is shown in figure 5.13 for LP compressor stages that consists of 6 stages of rotor-stator. Figure 5.14 shows the cascade of midline blade profile for IP compressor that consists of 9 stages. Figure 5.15 shows the cascade midline blade profile that have 10 stages rotor-stator in HP compressor. We can observed that LP compressor stages has lower flow deflection angle in compare to IP and HP compressor stages. In this case, HP has higher flow deflection angle than LP and IP compressor stages.

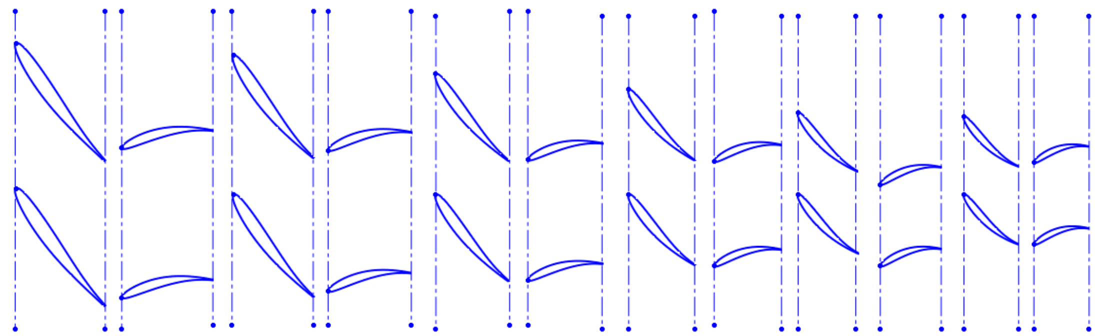

Figure 5.13. Cascade midline blade profile of LP compressor stages

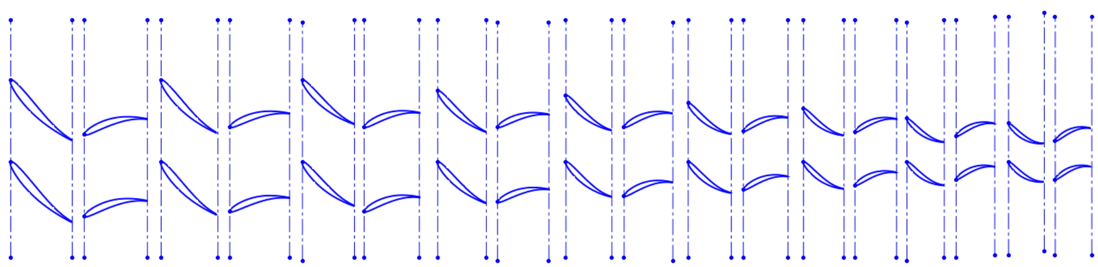

Figure 5.14. Cascade midline blade profile of IP compressor stages

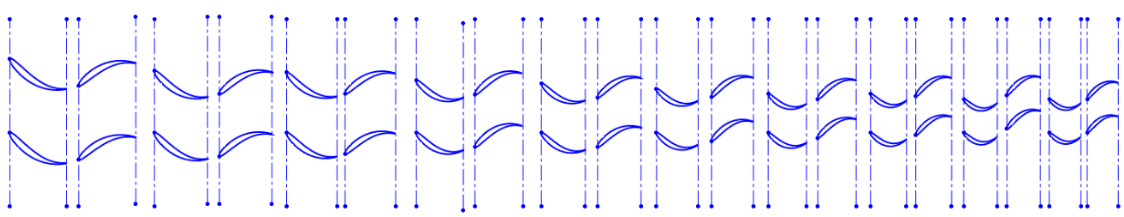

Figure 5.15. Cascade midline blade profile of HP compressor stages

We use Solidworks to create 3D modeling of multi-stage compressor as part of gas turbine engine. Figure 5.16 shows the Ultra High Efficiency Gas Turbine Section View. The 25 stages compressor is coupled with 3 combustion chamber and 6 turbine stages. A diffuser is placed between the outlet of HP compressor and the inlet of combustion chamber. We also can observed bleed valves between LP-IP compressor stages, IP-HP compressor stages, and at outlet of HP compressor stages. The bleed valve will serve as anti-surge valve and mainly open during start-up and shut-down of gas turbine operation and prevent the surge of compressor and making sure the operations are inside of compressor performance map.

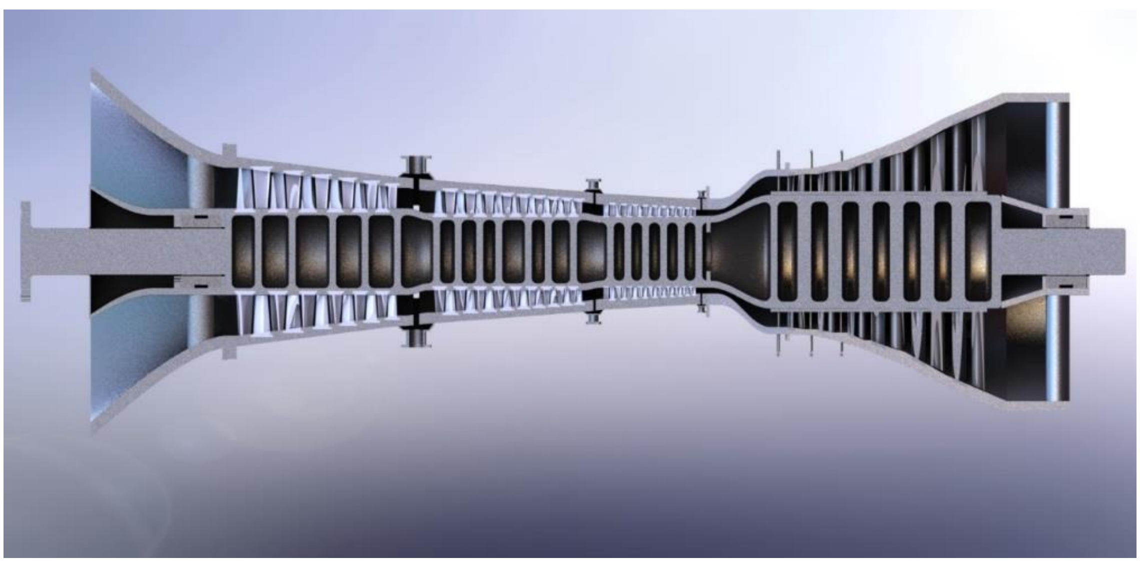

Figure 5.16. Section View of Ultra High Efficiency Gas Turbine (UHEGT)

Figure 5.17, figure 5.18, and figure 5.19 show the cascade view of LP, IP, and HP compressor stages, respectively, in 3D. It starts from the rotor row and followed by the stator row, etc. LP stages has lower flow deflection angle than IP and HP stages.

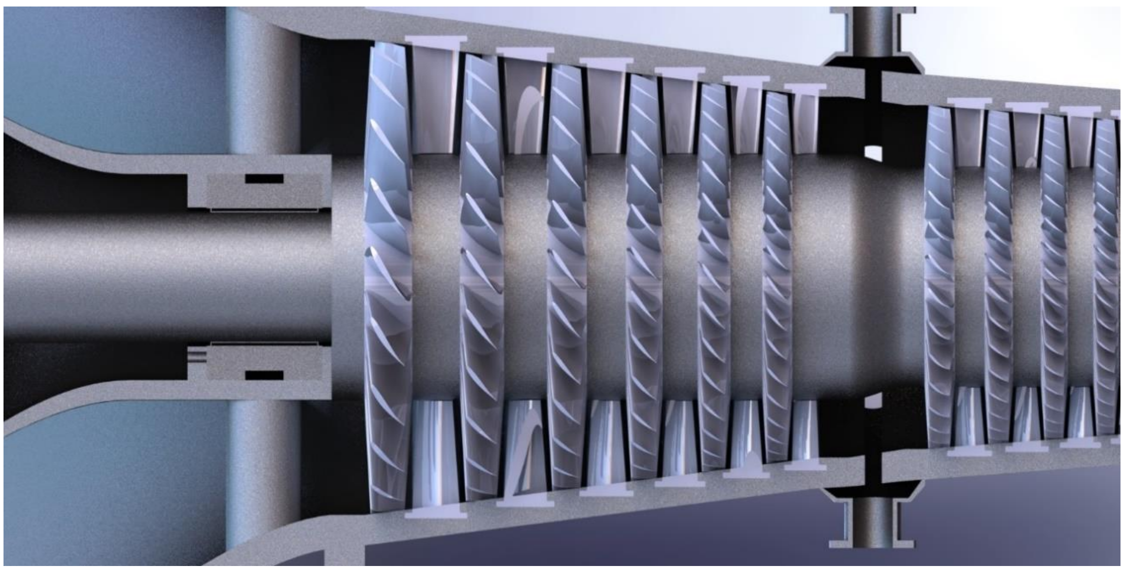

Figure 5.17. Cascade view of LP compressor stages

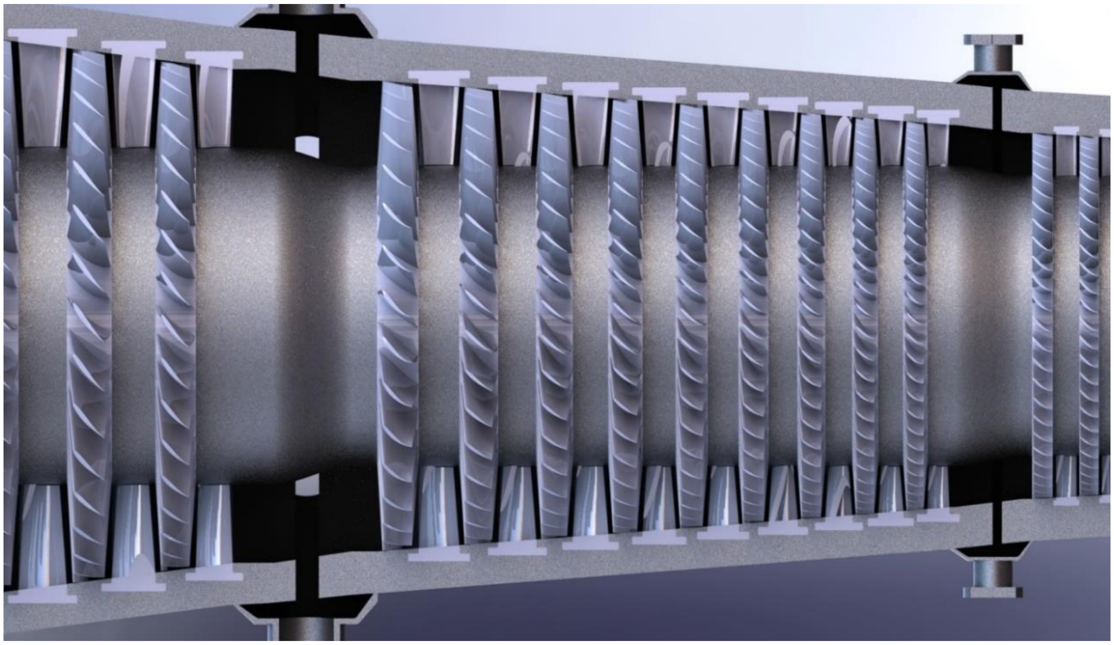

Figure 5.18. Cascade view of IP compressor stages

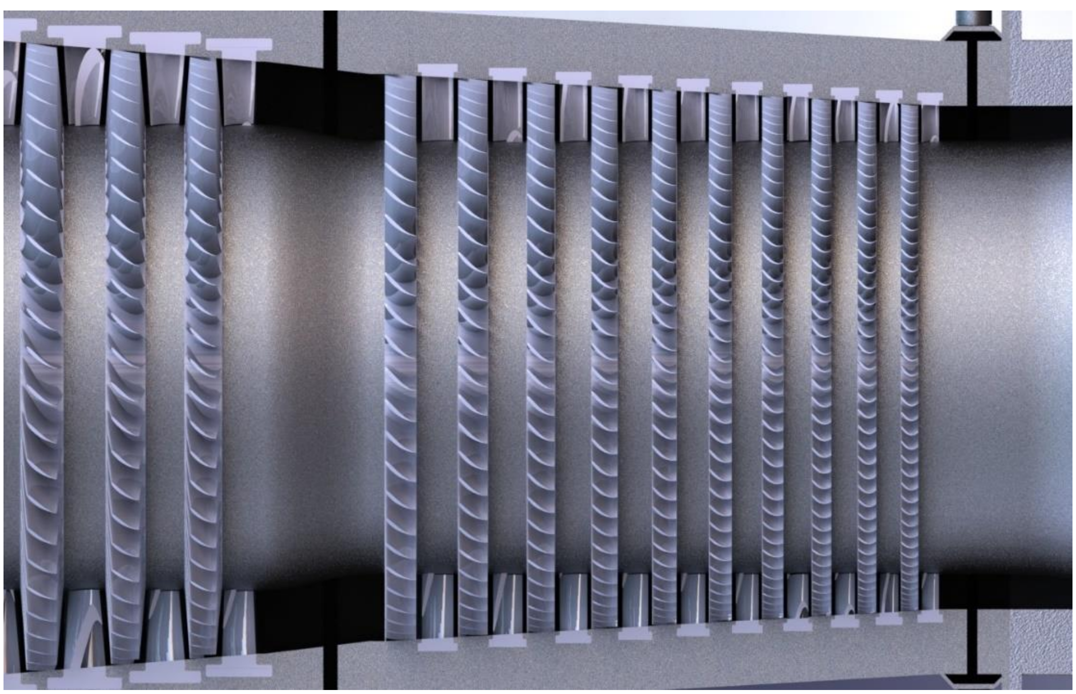

Figure 5.19. Cascade view of HP compressor stages

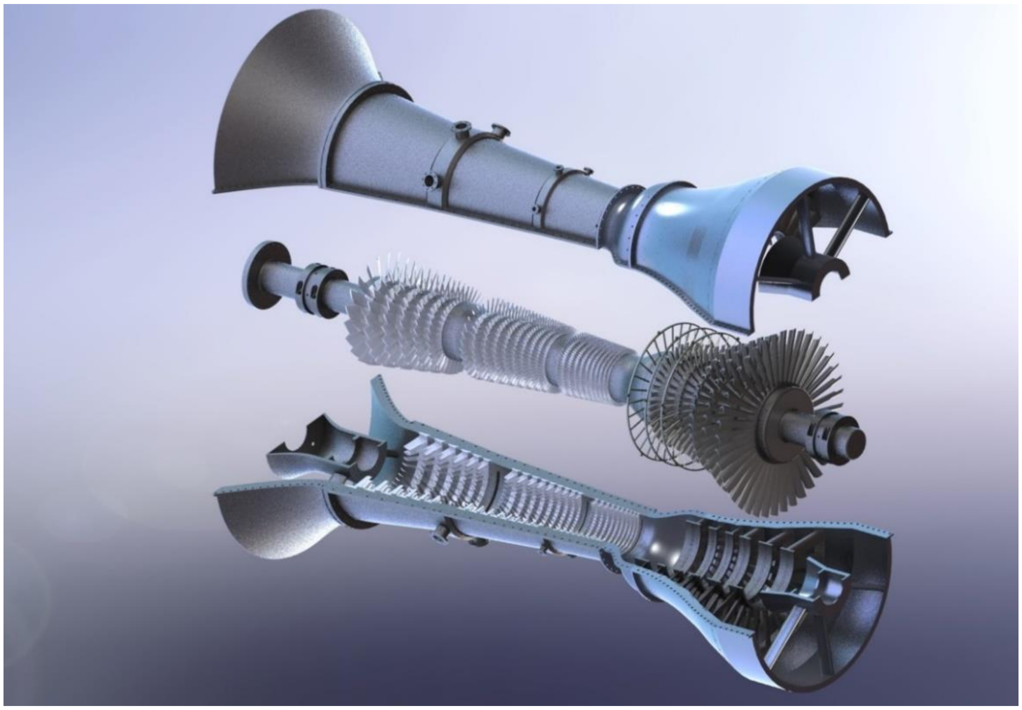

Figure 5.20. Exploded view of Ultra High Efficiency Gas Turbine (UHEGT)

Figure 5.20 shows the exploded view of Ultra High Efficiency Gas Turbine (UHEGT) and clearly shows the rotor shaft assembly. Figure 5.21 and figure 5.22 show the UHEGT section view of front side and rear side, respectively.

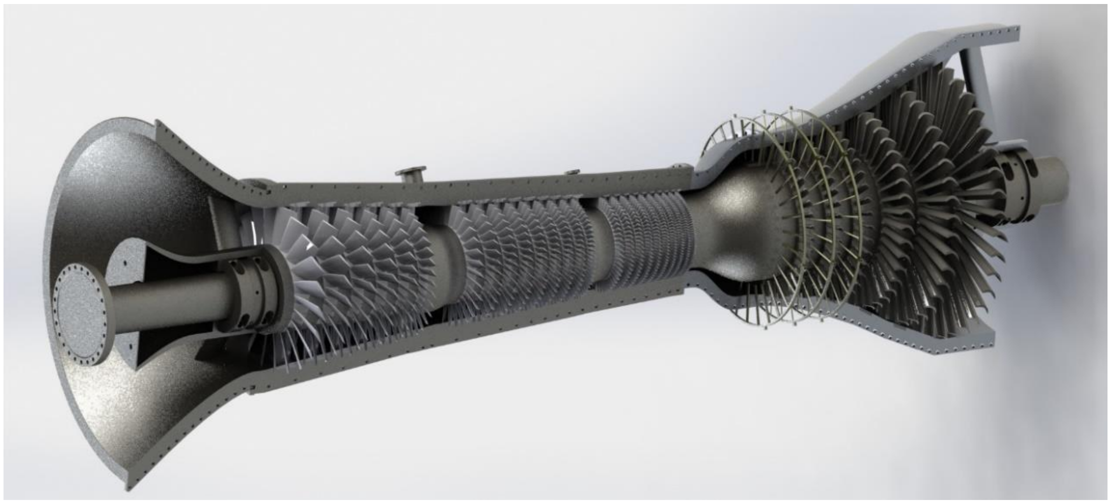

Figure 5.21. Section view (front side) of Ultra High Efficiency Gas Turbine (UHEGT)

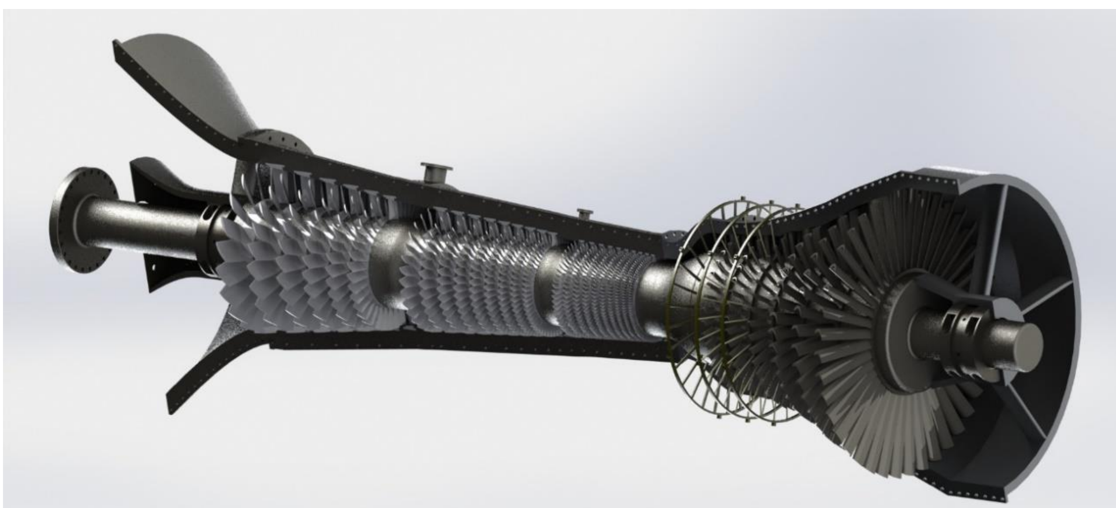

Figure 5.22. Section view (rear side) of Ultra High Efficiency Gas Turbine (UHEGT)

5.4. Compressor Design Performance Map

Compressor design performance maps are obtained from streamline curvature program by varying the compressor mass flow on design rotational speed (rpm). There are 3 compressor performance map which are for LP, IP, and HP stages. Each consists of isentropic efficiency as function of relative mass flow and total pressure ratio as function of relative mass flow. Generally speaking, reducing the mass flow will increase the isentropic efficiency and total pressure ratio. On the other hand, increasing the mass flow will resulting in a decrease of isentropic efficiency and total pressure ratio.

Starting from the design mass flow, reducing the mass flow will change the velocity diagram, and give higher flow deflection, stage load coefficient and total pressure ratio. By reducing the mass flow any further can lead to partial or total flow separation and compressor stall, moreover, when it reached the certain point, it will resulting in compressor surge. This point is known as surge limit.

On the other hand, if compressor back pressure is reduced, will also change the velocity diagram, and give negative flow deflection so that the axial velocity will be increasing. By reducing the back pressure any further can lead to have velocity equal to speed of sound, so the flow is choked. This point is known as choke limit.

Figure 5.23 and figure 5.24 show performance map of LP stages in term of isentropic efficiency and total pressure ratio versus compressor mass flow, respectively, at designed rpm. At designed mass flow, the isentropic efficiency = 92.49% and total pressure ratio = 3.6. We can observed that surge point is occurred in relative mass flow equal to 0.75 (with isentropic efficiency = 93.25% and total pressure ratio = 3.65). We also can observed that choke point is occurred in relative mass flow equal to 1.43 (with isentropic efficiency = 88.5% and total pressure ratio = 3.426).

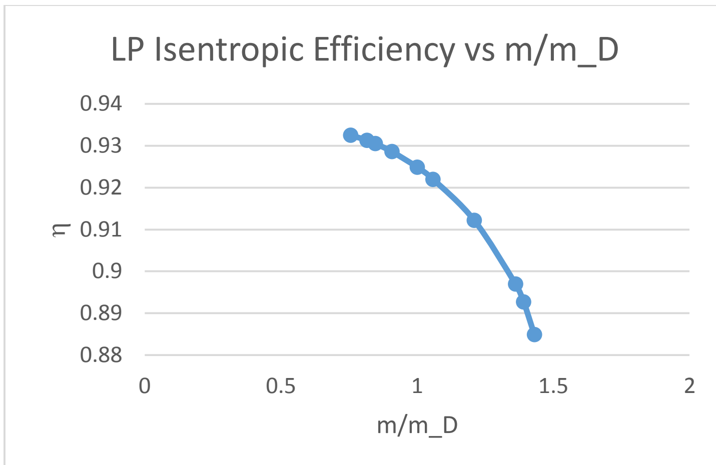

Figure 5.23. Isentropic efficiency as function of relative mass flow for LP compressor

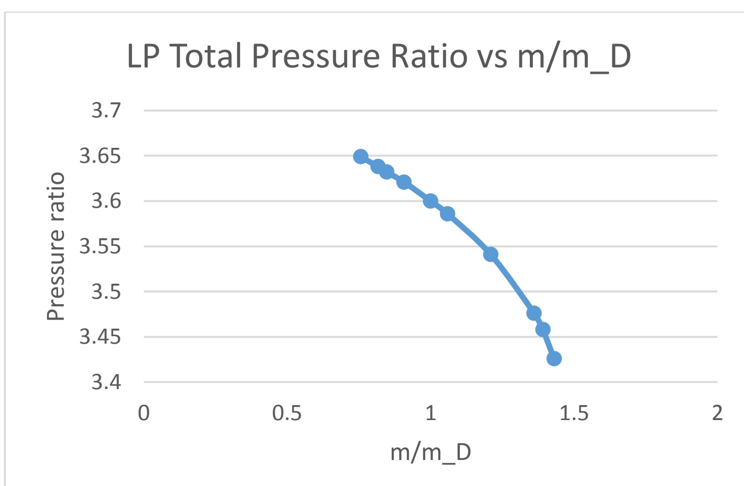

Figure 5.24. Pressure ratio as function of relative mass flow for LP compressor

Figure 5.25 and figure 5.26 show compressor performance map of IP stages at designed rpm. At designed mass flow, the isentropic efficiency = 91.03% and total pressure ratio = 3.4. We can observed that surge point is occurred in relative mass flow equal to 0.9 (with isentropic efficiency = 91.33% and total pressure ratio = 3.413). We also can observed that choke point is occurred in relative mass flow equal to 1.76 (with isentropic efficiency = 77.18% and total pressure ratio = 2.89).

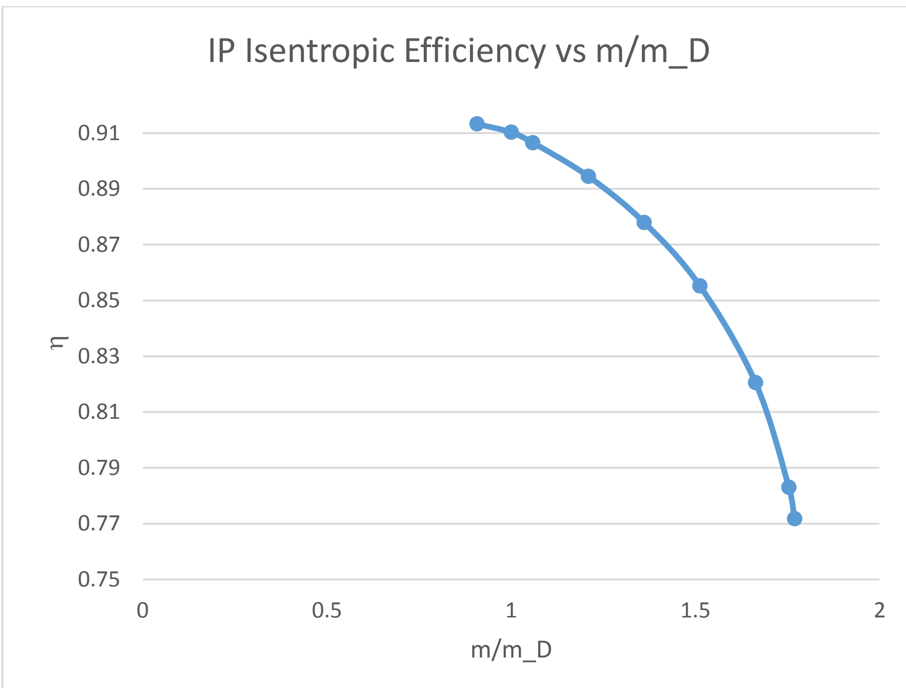

Figure 5.25. Isentropic efficiency as function of relative mass flow for IP compressor

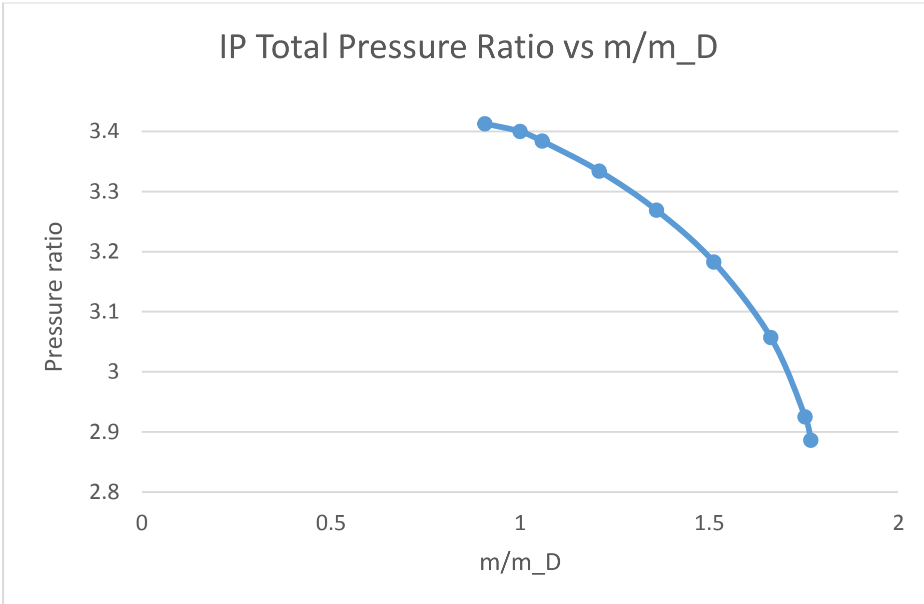

Figure 5.26. Pressure ratio as function of relative mass flow for IP compressor

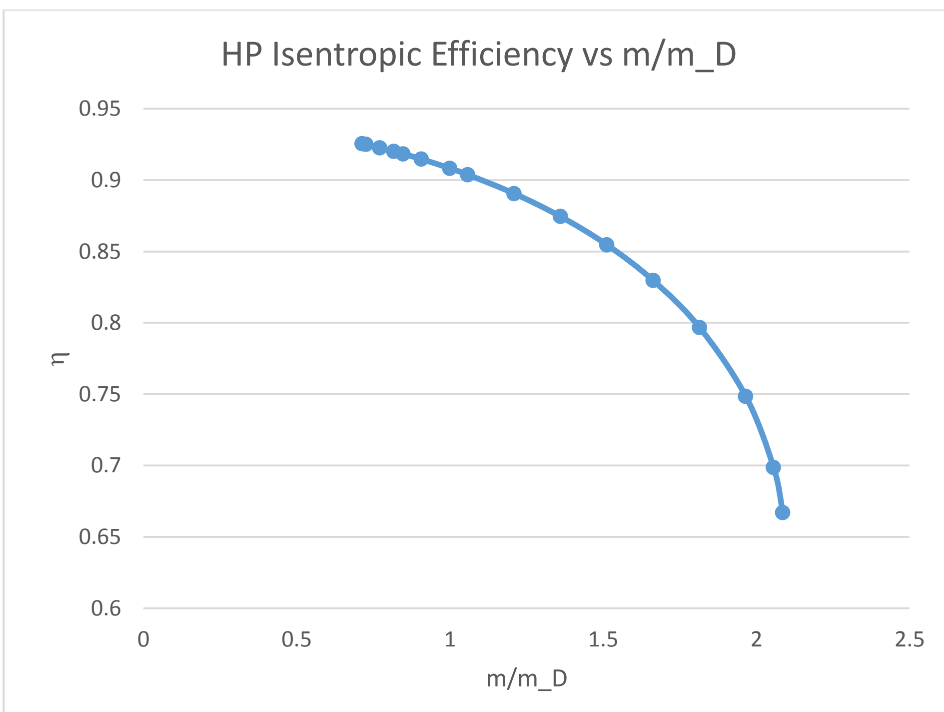

Figure 5.27. Isentropic efficiency as function of relative mass flow for HP compressor

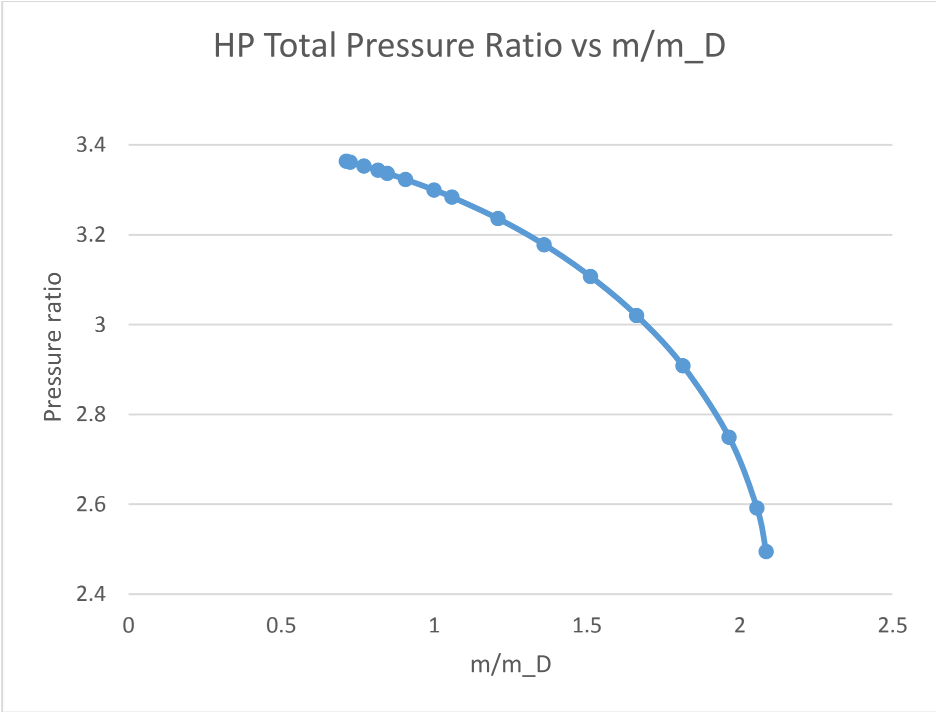

Figure 5.28. Pressure ratio as function of relative mass flow for HP compressor

Figure 5.27 and figure 5.28 show compressor performance map of HP stages at designed rpm. At designed mass flow, the isentropic efficiency = 90.83% and total pressure ratio = 3.3. We can observed that surge point is occurred in relative mass flow equal to 0.71 (with isentropic efficiency = 92.56% and total pressure ratio = 3.364). We also can observed that choke point is occurred in relative mass flow equal to 2.08 (with isentropic efficiency = 66.71% and total pressure ratio = 2.5).

5.5. Blade Pressure Distribution

CFD Package (Ansys Fluent) is used to generate the blade pressure distribution. Figure 5.29 shows the pressure distribution of compressor blade along the suction surface and pressure surface. The blade example is taken from IP stage rotor. This profile is based on design condition. The pressure distribution of pressure surface is higher than pressure distribution of suction surface. We can observed the stagnation point where the highest pressure is reached, occurred at leading edge of the blade. Shift from leading edge, the pressure is decrease until a point, and starting to increase until reached the trailing edge of the blade. This profile represent the blade pressure distribution of compressor blade.

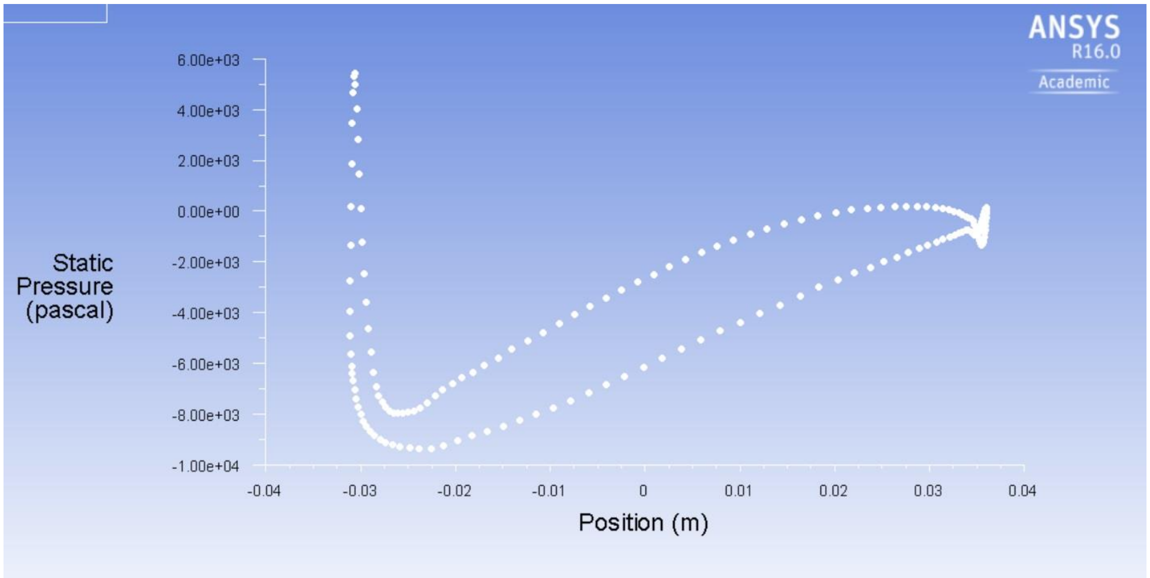

Figure 5.29. Blade pressure distribution of compressor blade (example from IP stage compressor)

6. CONCLUSIONS AND RECOMMENDATIONS

6.1 Conclusions

- The midline compressor blade calculation should be conducted as preliminary design and as input to streamline curvature program
- For design mode of streamline curvature method, enthalpy distribution and losses distribution (based on diffusion factor) is essential as input for rotor stations. On the other hand, exit angle distribution and losses distribution are used as input for stator stations. Basic geometry is also needed as the input data of streamline curvature method.
- Compressor performance map of design rpm can be generated from streamline curvature program.
- Streamline curvature method has the capability to generate streamline for LP, IP, and HP compressor stages.
- Streamline curvature method give lower deflection angle on LP and IP compressor stages, while higher deflection angle on HP stages. This issue may tend to flow separation in HP stages.

6.2 Recommendations

The studied showed that the streamline curvature method can be applied for LP, IP, and HP compressor stages. Further research is needed in order to obtain expected result and performance.

- In order to minimize flow separation in HP stages, we can decrease the overall HP stages pressure ratio to produce lower deflection angle of HP rotor blade
- Adding more stages in HP stages also can reduce HP stages pressure ratio (need compromising with exit blade height)
- Increasing the LP and IP stages pressure ratio also may be one alternative as HP pressure ratio can be decreased.

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